AIR-TO-AIR
HEAT EXCHANGERS

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by

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FOREWORD

The sponsors of this report, Canada Mortgage and Housing Corporation, Energy Mines and Resources, and the Housing and Urban Development Association of Canada are concerned about the conflicting requirements of reducing ventilation in housing in order to conserve energy on the one hand, and ensuring adequate ventilation for the health and safety of the residents on the other.

As the construction industry moves towards more energy efficient housing it is apparent that, in both new and retrofitted homes, significantly reduced natural air infiltration may present air quality problems, such as excessive moisture and contaminants. One possible solution, explored in this report, is controlled ventilation with heat recovery from the exhaust air.

"The Air-to-Air Heat Exchanger Study" is intended to be an introduction to the types of exchangers available, or soon to be available, in Canada. It describes the physical characteristics of the appliances in terms of various thermal, mechanical, electrical and acoustic criteria, and also partially analyzes the cost effectiveness of this type of device.

This report identifies criteria which may be used for determining the performance of units. As such, it provides guidance to designers, consumers and most importantly to standards writing organizations, who may establish performance measurement codes for heat recovery devices.

I. EXECUTIVE SUMMARY

This report has been prepared by The ECE Group Consulting Engineers and Ontario Research Foundation for the Technical Research Division of Canada Mortgage and Housing Corporation (CHMC), for the Department of Energy, Mines and Resources (EMR), and for the Housing and Urban Development Association of Canada (HUDAC). The purpose of this report is to provide them with an overview of the requirements for Air-to-Air Heat Recovery Devices for Canadian Housing.

The report provides a review of the Air-to-Air Heat Exchangers (ATAHE) which are presently available on the market or in an advanced stage of development and which could be made available in Canada. This review includes details of both physical and performance characteristics.

The performance requirements and physical characteristics of ATAHE for the Canadian market are determined. Consideration is given to: the range of ventilation rates required, frosting and icing protection, requirements for different regions in Canada, and various thermal, mechanical, electrical and acoustic and servicing criteria.

The operation and effectiveness of the various methods of frost protection are investigated. The interaction between ATAHE and other household equipment, as well as trouble free operation and maintenance requirements are also considered.

The cost effectiveness of operating ATAHE's with full frost protection and automatic shut-down are investigated. A step-by-step calculation method is described to determine simple payback periods for various capital costs and operating costs.

The use of ATAHE's to maintain a negative pressure within a house envelope to reduce condensation in walls and attics is investigated. This usage would only be feasible for very air-tight houses and the controls required would add to the cost of the ATAHE system. In Saskatchewan, researchers at NRC have noted problems in very air-tight houses tend to occur where air infiltrates the house around doors and windows. Depressurizing the house could add to these problems. Natural draft appliances would be affected by depressurization, and separate combustion air systems or induced draft fans would be required. An alternative to depressurization for condensation control is to use ATAHE's for humidity control using dry outside air.

Testing procedures for ATAHE's are presented to evaluate thermal effectiveness, defrosting and deicing effectiveness, cross contamination, sound levels and electrical power consumption. Test parameters suited to Canadian climates in addition to those established in other standards are included in the guidelines.

Test facility requirements for evaluation of ATAHE's in the residential capacity range are similar to those for larger heat recovery equipment. The differences in apparatus are the design air flow rates and the control of ambient environment conditions.

A survey was conducted to determine organizations which have the expertise, resources and facilities to evaluate the performance of test ATAHE's. Thirty-six (36) questionnaires were sent to appropriate organizations of which twenty-one have responded.

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INTRODUCTION

Recent increases in energy costs and improvements in building materials and construction methods have led to the creation of an artificial environment inside the typical Canadian House.

The air tight house envelope created to reduce the natural ventilation and infiltration can result in a high concentration of airborne contaminants within the house, and also in excessive humidity resulting in condensation and deterioration of the structural elements of a house due to moisture.

Most of these problems can be solved by including a controlled ventilation system in a house. To avoid an increase in energy consumption as a result of introducing such a system, an Air-to-Air Heat Exchanger (ATAHE) can be included in the ventilation system. An ATAHE will transfer energy from the exhaust air to the incoming outside air.

This study deals directly with all aspects of ATAHE's including feasibility, design, testing, operating modes and economics.

1.0 REVIEW OF EXISTING ATAHE SYSTEMS

In order to establish the present availability of ATAHE units built for residential purposes, over one hundred companies involved in the manufacture and supply of heat exchanging equipment were contacted. Of these, 16 identified their involvement in the manufacture or development of units suitable for residential applications. A list of these Companies can be found in Appendix I.

The companies were asked to respond with information relating to specific types of heat exchangers, including heat pipes, plate heat exchangers and heat wheels, which they might presently be manufacturing or distributing, and which would be applicable to our study. Since most of the ATAHE's built in the past have been for industrial applications which require flow rates of 470 L/s (1000 cfm) and more, the companies were also asked to supply us with information regarding the development of ATAHEs with capacity in the order of 50 L/s (106 cfm).

This rate was chosen as an adequate ventilation rate for the purpose of controlling humidity, odours and toxic gas build-up in the average Canadian home, as discussed in Section 2.

Most of the companies contacted are located in Canada and the United States, but letters were sent to manufacturers in Japan, England and the Scandinavian countries, where work in this field has been going on for many years.

Approximately 30% of the companies contacted responded to the survey. Some of the companies did not manufacture residential units but were ready to discuss the possibilities of producing such devices. Some companies offered specifications and cost estimates.

The interest and, in some cases, enthusiasm about getting involved in residential heat recovery expressed by the manufacturers is markedly different from the rather reluctant stand taken by some manufacturers at a CMHC Industry/Science Seminar on Controlled Ventilation

with Exhaust Air Heat Recovery for Canadian Housing, held in Ottawa in October 1978 (1). This change in attitude must be partially due to the increased level of public awareness as to the need for ventilation in homes to prevent the build-up of moisture, odours and toxic gases such as carbon monoxide and radon. The question of indoor air quality and mechanical ventilation has been given a large amount of coverage in recent years. The July 1979 Issue of the ASHRAE Journal (2) was devoted entirely to the problems of ventilation and infiltration. More recently, energy and engineering publications such as Energy Management Canada (3), Engineering Digest (4) and Canadian Renewable Energy News (5) have discussed the subject of mechanical ventilation and air-to-air heat exchangers, as have some less scientific magazines such as McLeans (6) Harrowsmith (7) and National Geographic (8).

This extensive coverage of the mechanical ventilation / energy recovery situation has resulted in a large number of companies becoming involved in the residential heat exchanger business.

The survey showed that in Canada there are approximately twelve manufacturers (at the time of this writing a number of these had not yet begun full scale production) involved in the production of ATAHE units for residential use and an even greater number in the United States.

The results of the survey are represented in Table 1. It can be seen that there are a large variety of ATAHE units for domestic purposes available on the market today. It is hoped that this report will help to set standards which will assist in the future development of ATAHEs for residential use.

Apart from private companies, a large amount of research in the ATAHE field is being done by the Government, universities and research institutions, although mostly for commercial and industrial applications.

TABLE 1: SUMMARY OF AIR-TO-AIR HEAT EXCHANGER SURVEY

S-3608
June 1981

COMPANY	TYPE OF ATAHE *						ORIGIN *			HEAT EXCHANGER CHARACTERISTICS				
	a	b	c	d	e	f	i	ii	iii	Airflow L/s (cfm)	Cost $	Maintenance	Adaptability	Dimensions
A		X					X		X	47 (100)	750	cleaning core	possible	14x18x69"
B			X					X		165 (350)	1000 approx.	washing matrix	possible	17x19x24"
E		X					X		X	47-236 (100-500)	500 - 1500	minimal	yes	16x17x48"
2		X						X		57-142 (120-300)	800	minimal	yes	16x13x35"
8		X						X		24-94 (50-200)	1000	minimal	roof inst.	40x48x100cm
11	X						X	X		47 up (100 up)	N.A.	general	yes	6x15x24"
24			X					X		47-111 (100-235)	700	cleaning core	yes	18x18x36"
33				X			X			any	N.A.	general	yes	6x6x144"
55		X					X		X	47-94 (100-200)	700	cleaning core	yes	14x14x60"
64		X						X	X	47 up (100 up)	100 and up	minimal	yes	3 ft^3
80		X						X		94-236 (200-500)	100 and up	minimal	yes	12x24x48"
83, 84		X						X	X	24-47 (50-100)	100 - 400	cleaning	yes	16x20x22"
89		X					X			28-85 (60-180)	795	none	yes	62"L.x19"D.
92		X					X		X	24-142 (50-300)	650 and up	oiling	yes	91x99x8cm
93	X							X		47 up (100 up)	250 and up	none	yes	6x9x12"
TOTAL	2	10	2	1	0	0	7	9	6					

* See Sample Questionnaire in Appendix I for descriptions.
Responding companies are indicated by ☐

INDEX TO TABLE 1

Company

A - The Air Changer Co. Ltd
B - Henlex Inc.
E - Conservation Energy Systems
2 - Indusco Sales
8 - Flakt Canada Ltd.
11 - Bry-Therm
24 - Ross & McNeilage Ltd.
33 - Trane Company of Canada Ltd.
55 - Des Champs Laboratories Inc.
64 - Exothermic Corp.
80 - United Air Specialists
84 - Advanced Idea Mechanics
89 - Enercon Industries Ltd.
92 - Atlantic Air Seal
93 - Q-Dot Corp.

Type of ATAHE

a Heat pipe
b Plate exchanger
c Heat wheel
d Coil loop
e Thermosiphon
f Other

Origin

i Manufactured in Canada
ii Available for importation
iii Presently in an advanced stage of development and which could be made available in Canada

2.0 PERFORMANCE REQUIREMENTS AND PHYSICAL CHARACTERISTICS FOR ATAHE SYSTEMS

2.1 MOTIVATION

In order to properly determine performance requirements and physical characteristics for Air-to-Air Heat Exchanger (ATAHE) systems, we should outline the two objectives behind air-to-air energy recovery as defined by the American Scciety of Heating, Refrigerating and Air-Conditioning Engineers (9):

Basic Objective: The reduction of a building's energy consumption and cost by transferring energy between supply and exhaust air streams.

Secondary Objective: The reduction of size and capital cost of the supporting utility equipment of the building.

The ATAHE units under consideration will be required to cause a reduction in the annual energy consumption and annual energy cost of a house (including the operation

and maintenance costs of the heat exchanger), when compared with a house with the same rate of ventilation by natural means, or by mechnaical means without an ATAHE. If this requirement is not met, an ATAHE will not be economical, despite the fact that it may save energy.

Careful attention in the design and construction of a new house will result in a comparatively large reduction in heating load compared to currently existing houses. The use of an ATAHE with mechanical ventilation (instead of relying solely on natural ventilation) might be expected to reduce a single family dwelling's heating load by a further 1700 W (7000 Btu/hr).

This heat load is based on the calculations determined in section 4.4. At 50 L/S ventilation rate, the heat will amount to 2.4 KW. Assuming that 70% of this is recoverable, then the reduction in heat load becomes 70% of 2.4 KW = 1.7 KW.

Currently residential furnaces are typically available in model size increments of 3.5 KW (12,000 Btu/hr). We might expect, therefore, that on this basis 50% of houses (1.7 KW ÷ 3.5 KW) might have their furnace downsized.

The above presumptions are based however on the ATAHE operating effectively at the design temperature, i.e. that it does not shut down because of frosting up.

2.2 ATAHE PERFORMANCE AND PHYSICAL CHARACTERISTICS

The performance of an ATAHE is usually expressed in terms of its effectiveness in transferring:

(1) sensible heat (dry-bulb temperature)

(2) total heat (enthalpy)

The effectiveness, ε, of a heat exchanger is defined as follows:

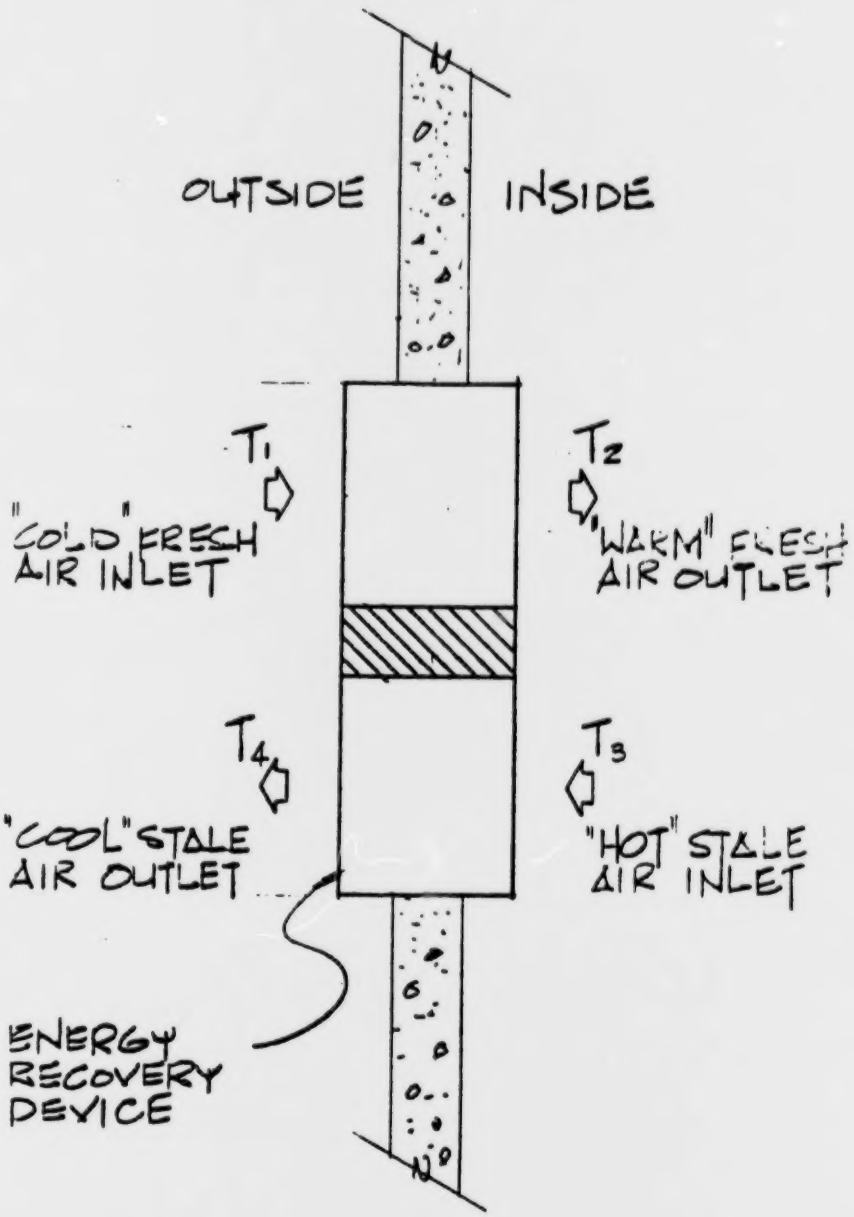
$$\varepsilon = \frac{\text{actual transfer for the given device}}{\text{maximum possible transfer between the airstreams}}$$

Referring to Figure 1, the sensible heat effectiveness, ε_{sh} can be defined as:

$$\varepsilon_{sh} = \frac{W_2 Cp_2 T_2 - W_1 Cp_1 T_1}{W_3 Cp_3 T_3 - W_1 Cp_1 T_1}$$ in terms of the heat gain of the supply air.

$$\varepsilon_{sh} = \frac{W_3 Cp_3 T_3 - W_4 Cp_4 T_4}{W_3 Cp_3 T_3 - W_1 Cp_1 T_1}$$ in terms of the heat loss of the exhaust air.

where: W = mass flow rate
Cp= specific heat
T = dry bulb temperature



FLOW of AIR THROUGH HEAT EXCHANGER
(WINTER CONDITIONS)

Detail No.
FIGURE 1

Standard N °

Standard ¼" riser

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For balanced flow, with negligible cross flow leakage and the temperature ranges we are concerned with, we can assume:

$$W_1 = W_2 = W_3 = W_4$$

and

$$Cp_1 = Cp_2 = Cp_3 = Cp_4$$

Unless there are extraneous heat losses or gains, we should expect:

$$\varepsilon_{sh} = \frac{T_2 - T_1}{T_3 - T_1} = \frac{T_3 - T_4}{T_3 - T_1}$$

Similarly, the total heat effectiveness, ε_{th}, can be defined as:

$$\varepsilon_{th} = \frac{h_2 - h_1}{h_3 - h_1} = \frac{h_3 - h_4}{h_3 - h_1}$$

where h = enthalpy

A third method of describing the effectiveness of an ATAHE is to measure the ability of the heat exchanger to transfer latent heat (humidity ratio). The latent heat effectiveness can then be defined as:

$$\varepsilon_{1h} = \frac{X_2 - X_1}{X_3 - X_1} = \frac{X_3 - X_4}{X_3 - X_1}$$

where X = humidity ratio

The latent heat effectiveness is relevant only in special cases where the ventilation rate to control humidity levels is over-shadowed by some ventilation requirements. An example of such a situation is in an area such as Elliot Lake, where control of radon levels may be the main ventilation requirement, and a heat exchanger with no latent heat recovery would result in very low indoor humidity levels.

The performance of any heat recovery system depends upon the following factors:

a) The difference in dry-bulb temperature, humidity ratio or enthalpy between supply and exhaust air streams;

b) The supply and exhaust mass flow rates and specific heats;

c) The effectiveness of the particular heat transfer device;

d) The extra energy input required to operate the heat recovery device;

e) The fan or pump energy absorbed as heat by the heat transfer device and which can enhance or detract from the performance.

The various types of ATAHEs employ different methods to transfer energy between the exhaust and supply air streams. These differences are of primary importance in determining the physical and operational characteristics of ATAHEs as well as the type of energy recovery capabilities. A brief description of the five major types of heat exchangers will be given in the following:

1. Heat Pipes;

2. Plate Heat Exchangers;

3. Heat Wheels;

4. Coil Energy Recovery Loops;

5. Thermosiphon Heat Exchangers.

The first three require that the supply and exhaust air streams be adjacent to one another for energy recovery to take place. Coil energy recovery loops have the advantage of being able to transfer heat through piping between supply and exhaust ducts located remotely from each other. Thermosiphon heat exchangers can be designed to transfer heat in either situation.

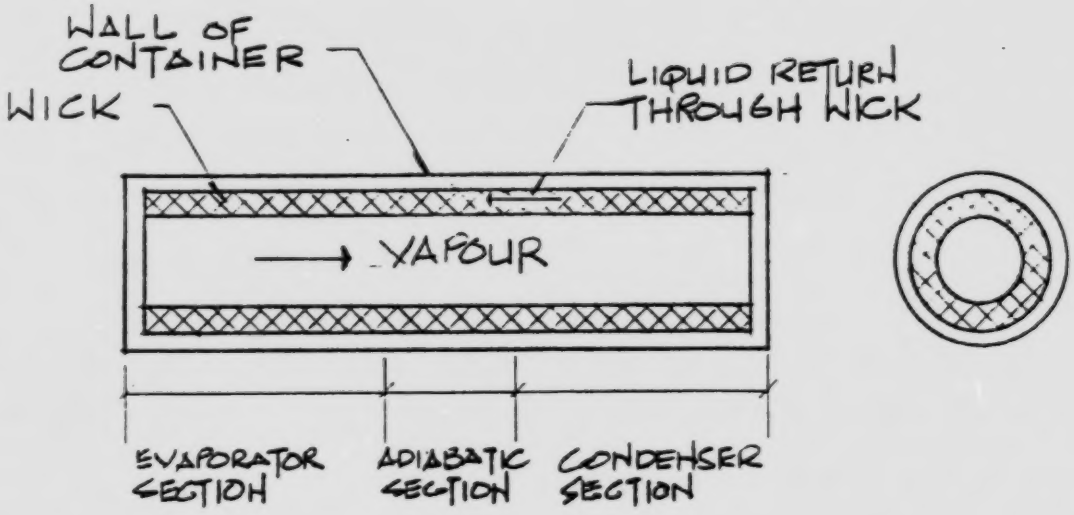
2.2.1 The Heat Exchangers which require adjacent air streams are:

1) HEAT PIPES

A heat pipe is a tube with a capillary wick structure, filled with a refrigerant and permanently sealed. It operates on a continuous condensation/evaporation cycle. One end of the tube causes the liquid to vaporize. The vapour then moves to the cold end of the tube where it condenses. A wick, constructed for example from a

few layers of fine gauze fixed to the inside surface, returns the condensate to the evaporator section by capillary action. Since the latent heat of evaporation is large, considerable quantities of heat can be transported with a very small temperature difference from end to end. Thus the structure will have a high effective thermal conductance. The basic construction of the heat pipe is shown in Figure 2. By a suitable choice of working fluid it is possible to construct heat pipes which would have a thermal conductivity several hundred times that of a solid metal rod.

A heat pipe exchanger consists of a bundle of externally finned tubes which are made up as individual heat pipes of the type described above. The heat pipe evaporation/condensation cycle causes the transfer of heat from the "evaporators", located in the duct carrying the air stream from which heat is to be recovered, to the "condensers" in the adjacent duct carrying the air which is to be heated, as shown in Figure 3. In the heat pipe exchanger, the tube bundle may be horizontal or vertical since the condensate return does not rely solely on the force of gravity, though the angle of the heat pipes may be adjusted as a



LONGITUDINAL SECTION

CROSS SECTION

BASIC HEAT PIPE CONFIGURATION

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FIGURE 2

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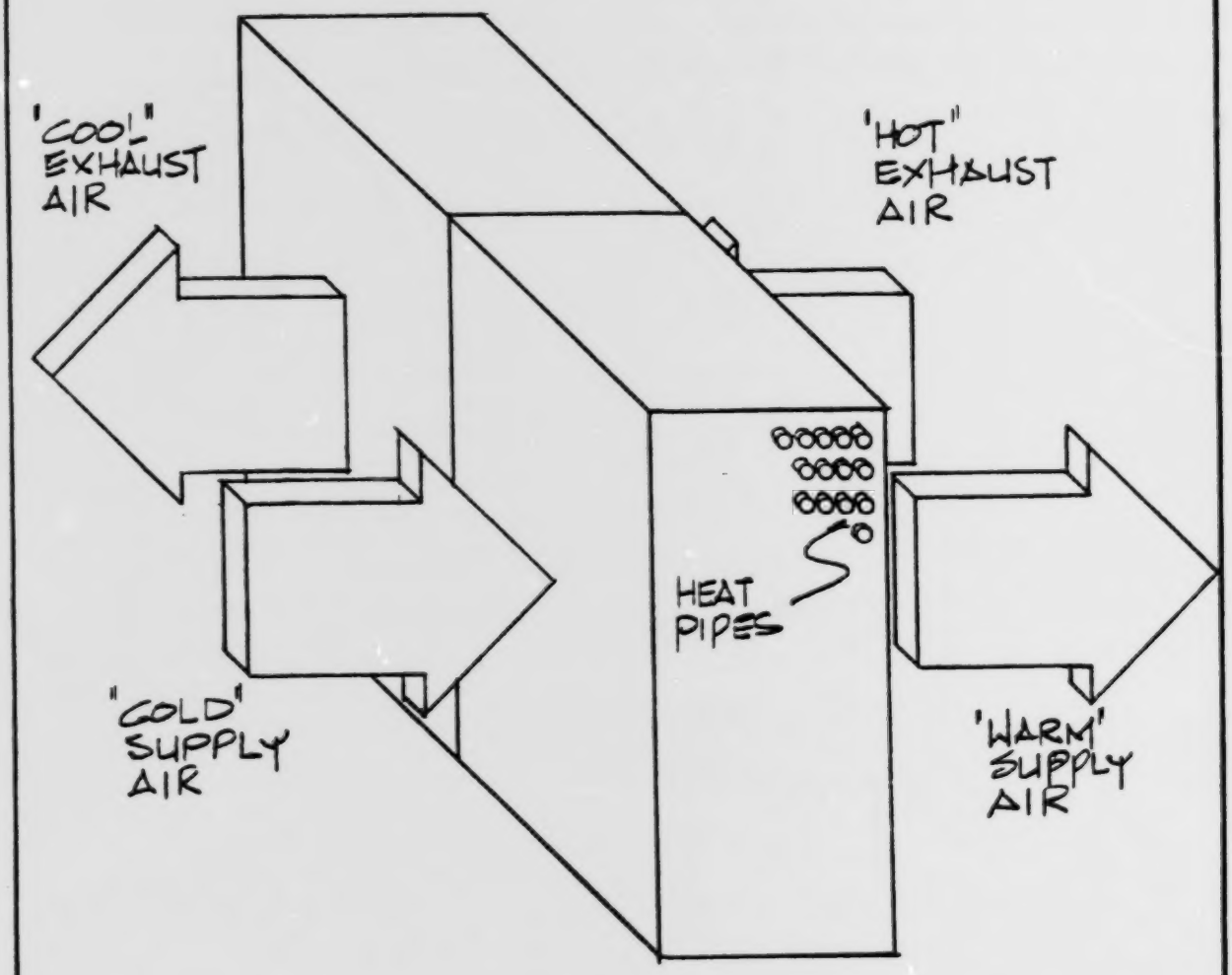


HEAT PIPE HEAT EXCHANGER

Detail No.

FIGURE 3

means of controlling the heat transport. This is a useful feature in air conditioning applications.

Features of the heat pipe exchanger which are attractive are:

(a) no moving parts and no external power requirements, implying high reliability;
(b) no cross contamination between the air streams;
(c) The heat pipe is fully reversible, i.e., heat can be transferred in either direction.

Heat pipe heat exchangers usually have a sensible effectiveness of 60 - 70%.

2) PLATE HEAT EXCHANGERS

This type of ATAHE is the most common and has the simplest construction. It consists of parallel plates forming channels inside which the warm and cold air streams are drawn past each other without cross-contamination. The plates are made of heat conducting metal such as aluminum, or for domestic application, plastic material. Although plastic

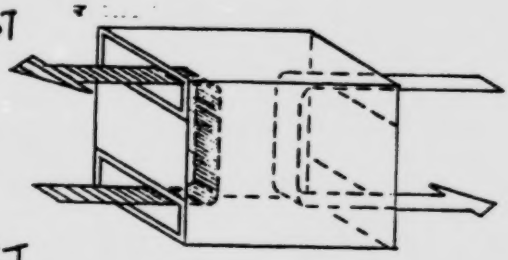
is not a good conducter, it does have redeeming characteristics. In one design, polyethylene film is used, and because of its low cost the designer could afford to use a large surface area to achieve the heat transfer. Several designs incorporate multiple layers of hollow core polyethylene extruded sheet. Low material cost, simplicity of fabrication and low weight are beneficial trade offs against the lower conductivity than that of a metal heat exchanger.

The plate may be arranged either to cause a counterflow between the two air streams or a crossflow. The former is more efficient, but because mainfolding is more difficult, is more expensive than the crossflow arrangement. Some of the more common arrangements are illustrated in Figure 4.

As with the heat pipe system, the plate heat exchanger offers high reliability due to its lack of moving parts or need for external power requirements. Two types of plate exchangers are available, the pure- plate heat exchanger consisting of primary transfer surface only and plate-fin exchanger, consisting of plates and interconnecting fins.

The sensible heat effectiveness of this type of heat exchanger is 50 - 80% for the pure-plate exchanger and 40 - 60% for the plate-fin type.

PURE-PLATE COUNTERFLOW



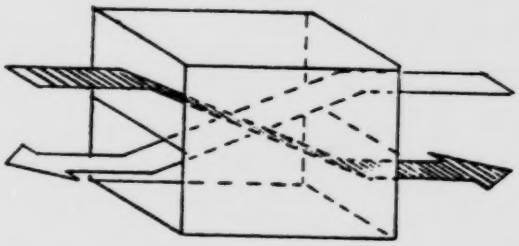


PLATE-FIN CROSSFLOW

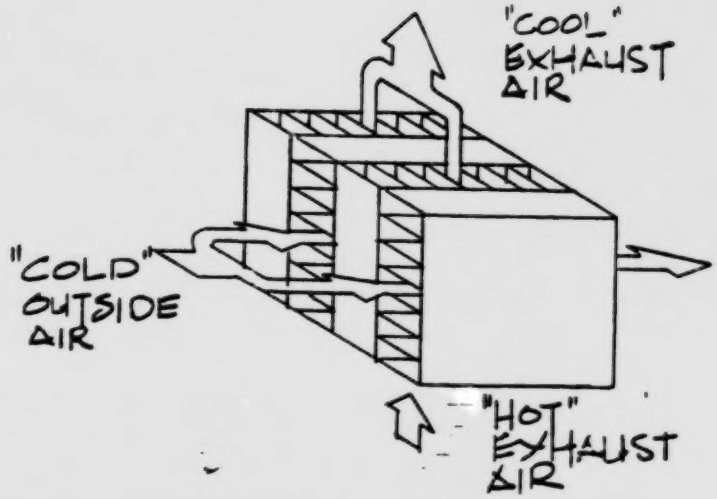


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FIGURE 4

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3) HEAT WHEELS

A heat wheel is a rotary heat exchanger with a high thermal inertia core driven by an electric motor (Figure 5). The core is in the form of a porous cylinder made for example of aluminum honeycomb. This rotates on an axis parallel and between the exhaust and supply air streams. Heat is absorbed by the portion of the core passing through the exhaust stream and is subsequently given up to the intake air stream, thus warming it.

Two types of heat wheels are available, one which transfers sensible heat only and the other which transfers both sensible and latent heat.

Unlike the other heat exchangers discussed in this report, heat wheels have the disadvantage that they allow cross-contamination between the two air stream. This cannot be avoided, since the heat exchanger medium is occupied alternately by both airstreams. This carryover though can be limited to about 1% for industrial ventilation applications by the use of seals. In a residential situation the cross-contamination could be as high as 5% and not present a significant problem.

Heat wheels typically have an effectivenss of 60 - 80% for sensible heat and 20 - 60% for latent heat.

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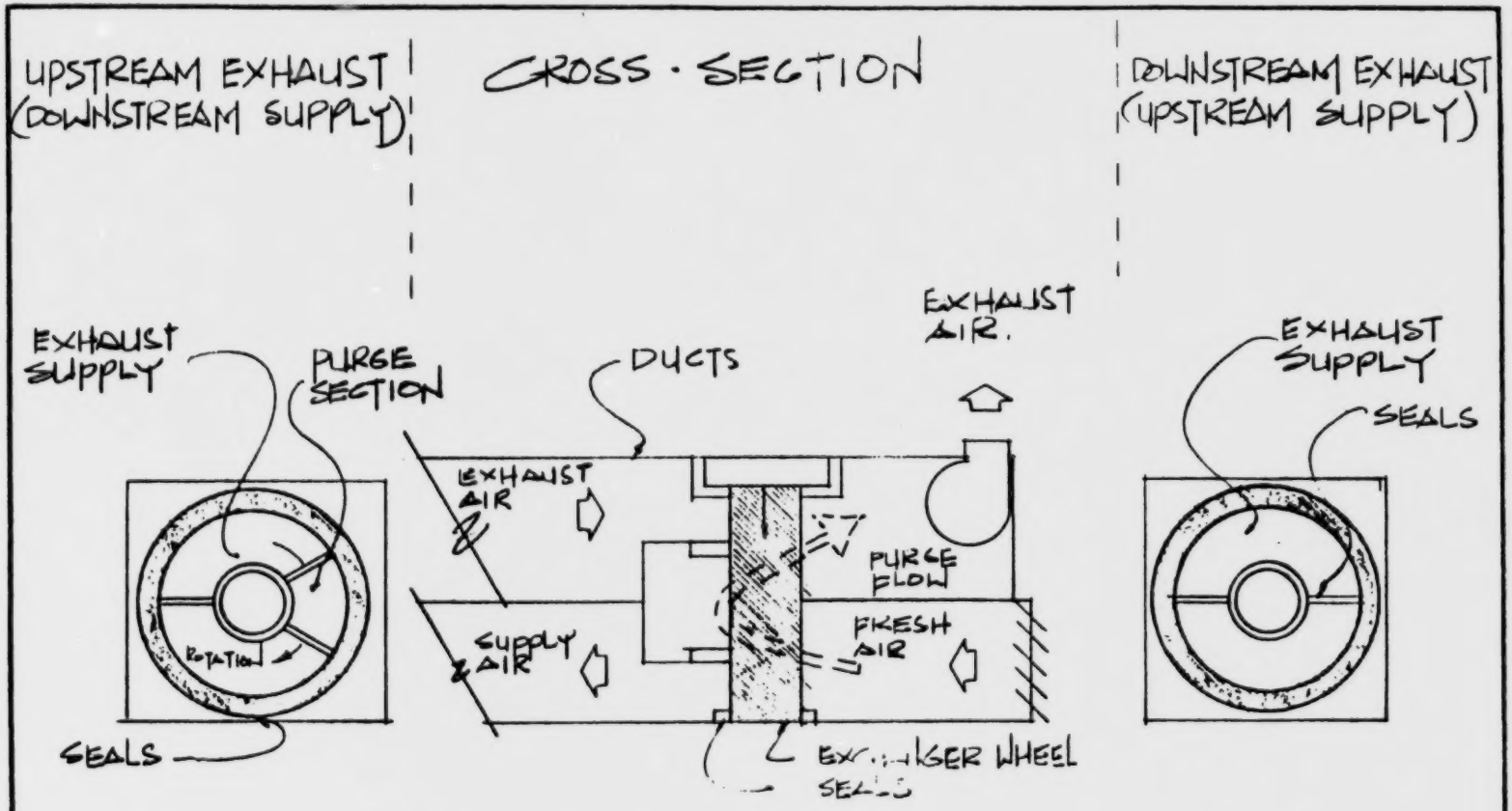


THERMAL WHEEL HEAT EXCHANGER SECTIONS

Detail No.
FIGURE 5

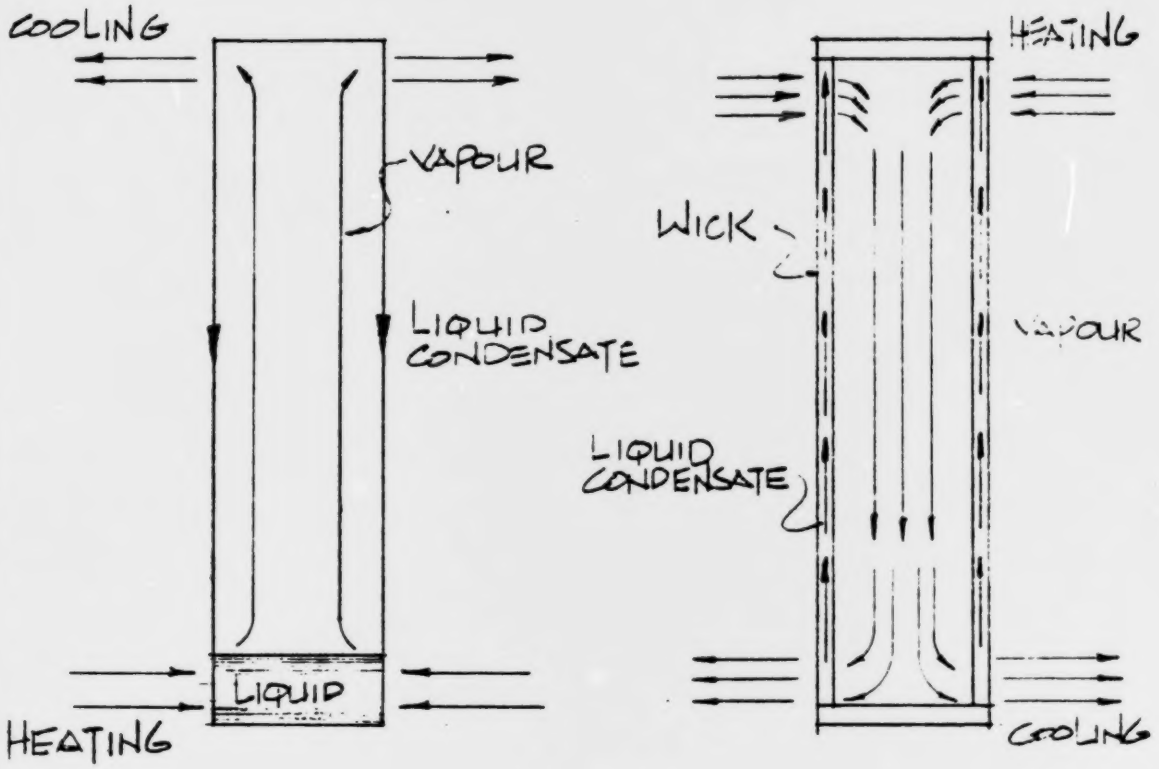
4) THERMOSIPHON HEAT EXCHANGERS

The thermosiphon is similar in some respects to a heat pipe with the main difference being that where the latter depends on a wick to return the condensate to the evaporator, in the thermosiphon the condensate is returned to the hot end by gravity (Figure 6). This has the advantage of making the thermosiphon substantially easier and cheaper to build, but also the disadvantage of having to locate the evaporator below the condenser in order to effect energy transfer. If installed for winter operation, it cannot also be effective for a summer operating condition.

Thermosiphon heat exchangers can be constructed like the heat pipe heat exchangers described above consisting of bundles of sealed tube thermosiphons lying across adjacent air ducts.

The sensible heat effectiveness of most thermosiphons is similar to that for heat pipes, about 60 - 70%.

COMPARISON OF THERMOSIPHON & HEAT PIPE



THERMOSIPHON

HEAT PIPE

Standard Number

Standard Name

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Detail No.
FIGURE 6

2.2.2 The Heat Exchangers which can be applied to non-adjacent airstreams are:

1) COIL ENERGY RECOVERY LOOP

This heat recovery system consists of separate finned tube coils placed in the exhaust stream and in the outside air make-up stream. A liquid is circulated through the coil tubes and transfers sensible heat only from the exhaust air to the make-up air.

The main advantage that this system has over the previous three is its high degree of flexibility, as it can accommodate remotely located supply and exhaust ducts and would thus be well suited for retrofitting. This system requires a pump and set of valves, piping, and expansion tanks, and the cost of installation and maintenance may restrict its use to commercial and industrial applications.

This system is seasonally reversible, and since heat is transferred through piping, does not exhibit any cross-contamination.

The sensible effectiveness for a coil loop system is about 50 - 60%.

II) THERMOSIPHON HEAT EXCHANGERS

The coil loop thermosiphon is constructed like the coil energy recovery loop, without the need for a pump but with condenser and evaporator coils.

This system has flexibility similar to the coil loop system without the need for external power supply, but is limited to shorter distances and is not reversible.

A summary of the performance and physical characteristics of the five types of Air-to-Air Heat Exchangers is given in Table 2.

2.3 OPTIMUM VENTILATION RATES

The question of optimum and minimum ventilation rates received much attention in recent years. Ventilation is required in order to maintain an acceptable level of

TABLE 2 ATAHE CHARACTERISTICS.

	HEAT PIPE	PLATE	HEAT WHEEL	COIL	THERMOSIPHON
Sensible Heat Effectiveness(%)	60-70	50-80	60-80	50-60	60-70
Latent Heat Transfer Ability	NO	NO*	YES	NO	NO
Extra Power Requirements	None	None	Motor	Pump	None
Cross Contamination	None	None	1%	None	None
Maintenance Required	1	2	3	3	1
Frost Control Available	2	3	1	3	2
First Cost 47 L/s (100 cfm)	3	1	2	3	3

1 = Best Rating
2 = Average Rating
3 = Poor Rating

* Latent Heat Transfer available in some cases.

air quality within the dwelling. This requirement usually results from the need to:

- maintain proper oxygen and carbon dioxide levels;
- control the concentration of contaminants that may produce objectionable odours or toxic effects;
- control the humidity levels within the building;
- provide adequate air for any combustion device within the house.

Fresh outside air contains about 21% O2 and 0.03% CO2 on a volume basis (11). For prolonged exposure, a minimum concentration of 16% O2 and a maximum concentration of 0.5% CO2 is acceptable. A ventilation rate of 1.4 L/s (3 cfm) per person at rest would be required to maintain the O2 and CO2 levels between these limits.

In 1973, ASHRAE (American Society of Heating, Refrigerating and Air-Conditioning Engineers) issued Standard 62-73 for Natural and Mechanical Ventilation (12). This standard defined minimum and recommended ventilation air quantities required to maintain acceptable O2 and CO2 levels. It does not attempt to

specify the air quantities required for the control of humidity or contaminants.

A sample of the ventilation requirements for private dwellings from ASHRAE Standard 62-73 in shown in Table 3.

In 1975, in response to demands for energy efficient buildings, ASHRAE published Standard 90-75, "Energy Conservation in New Building Design" (13). This standard has resulted in a conflict with Standard 62-73 by effectively deleting the recommended column in 62-73 and suggesting that the minimum column be used when designing new buildings (38,39,40,41).

To resolve this conflict, Standard 62-73 has been revised, and some of the changes are summarized in Table 4.

Apart from the control of O2 and CO2 levels, ventilation is required to control other indoor contaminants such as:

- odours;
- particulates;

TABLE 3. ASHRAE STANDARD 62-73
VENTILATION REQUIREMENTS FOR SINGLE UNIT DWELLINGS.

	Required Ventilation Air L/s-person (cfm/person)	
	Minimum	Recommended
General Living Areas, Bedrooms	2.4 (5)	3.3-4.7 (7-10)
Kitchens	9.4 (20)*	14-24 (30-50)*
Baths, Toilet Rooms	9.4 (20)*	14-24 (30-50)*
Basement, Utility rooms	2.4 (5)	2.4 (5)

* Installed capacity for intermittent use.

Comments:
These requirements are for 100% outdoor air.
Outdoor air requirements may be reduced to 33% except where recirculation is prohibited.
In no case should the outdoor air quantity fall below 2.4 L/s-person (5 cfm/person).

TABLE 4. COMPARISON OF VENTILATION REQUIREMENTS BETWEEN ASHRAE STANDARD 62-81

	ASHRAE 62-73	ASHRAE 62-81
Outdoor air quality	acceptable criteria spcified	acceptable criteria more precisely specified
Ventilation air	treated circulated air allowable	treatment more precisely specified
Mechanical Ventilation	reduced rates allowable	methods for intermittent ventilation described
Primary Ventilation Criteria	cfm/person, min. and recommended values	required values of cfm/person specified for smoking and non-smoking areas
Indoor air quality	not specified	alternate performance method specified
Compliance	not specified	methods of calculation and measurement specified

Source: Ventilation, Health & Energy Consumption: A Status Report, J.E. Woods (14).

- combustion products;
- formaldehyde and other toxins;
- radon, and
- humidity.

In several recent studies concerned with air quality (15,16,17) the ventilation rates required to control these contaminants have been investigated.

The ventilation rates required to control body odours and tobacco smoke vary from 4.7 to 14.2 L/s (10 to 30 cfm) per person for the majority of situations.

Laboratory studies of CO and NO2 concentrations in a test room with a gas oven operating for one hour at 180°C (350°F) showed that a ventilation rate of from 24 to 47 L/s (50 to 100 cfm) was required to keep the toxic gas concentrations within the limits established by air quality health standards (18).

Measurement of radon levels in buildings has shown that indoor radon concentrations can be several orders of magnitude higher than outside concentrations (19). Results also indicate that an air exchange rate of

approximately 0.5 air changes per hour (ach) may be required in order to maintain radon concentrations below the maximum permissible concentration given by health guidelines. This value is also the accepted minimum ventilation rate standard for residential buildings recommended in the Nordic Countries (Denmark, Finland, Norway and Sweden).

The control of indoor humidity is one of the most important requirements for any ventilation system. The amount of indoor moisture production can vary between 2.3 to 23 kg (5 to 50 lbs.) per day. It is estimated that the average Canadian family of four produces 6.8 to 11.3 kg (15 to 25 lbs.) of moisture per day (20). If this moisture were not removed during cold weather, condensation would occur on windows, walls and in unheated crawl spaces, where it can cause deterioration to the insulation and the building structure. In the majority of residential applications humidity control determines the ventilation rate. An ATAHE unit can be used to lower the humidity level inside a house by introducing dry, outdoor air with reduced penalty in heating energy.

The results of a large number of studies on the indoor humidity problem, indicate that a continuous ventilation rate of from 28 to 47 L/s (60 to 100 cfm), depending on individual living habits, is required to control the moisture build-up in a typical Canadian home.

A summary of the various ventilation requirements is shown in Table 5.

From this Table it can be seen that the largest ventilation rate required is 47 L/s (100 cfm) for both control of combustion products and humidity control.

Since this rate would also satisfy the requirements for the control of the other contaminants and maintain the proper O2 and CO2 levels, a ventilation rate of 50 L/s (106 cfm) was chosen for this study.

The value of 50 L/s also appears in the National Building Code, as well as other provincial and municipal codes throughout Canada. Section 9.33 of the 1980 Edition of National Building Code of Canada (21) states:

TABLE 5. SUMMARY OF VENTILATION REQUIREMENTS

VENTILATION PURPOSE	VENTILATION RATES REQUIRED		REFERENCES	COMMENTS
	L/s (CFM)	ACH		
Metabolic Processes	1.4-2.4 (3-5)		11,12	per person
Odour/Smoke Control	5 - 14 (10-30)		15	per person
Control of Combustion Products	24-47 (50-100)		17,18	For gas stove in a laboratory
Formaldehyde Control		0.3	17	From European indoor standard
Radon Control		0.5	17,19	Given by Health Guidelines
Humidity Control	28-47 (60-100)		16,17,20	For average family

In order to relate the ventilation requirements in terms of ach to air flowrate, assume an average size house with 140 square metres (1500 ft. sq.) floor area and 2.3 metres (7.5 ft.) ceilings:

-Volume is 320 cubic metres (11250 ft^3)
-1 ach = 88 litres/second (187.5 cfm)

Therefore a 0.5 ach requirement for radon control would equal a flowrate of 44 L/s (94 cfm).

"Rooms and spaces in buildings of residential occupancy shall be ventilated by natural means ... or by mechanical means ... except that where a dwelling unit is heated with other than fuel-fired equipment within the dwelling unit, a mechanical exhaust system of 1 or more fans or blowers having a total capacity of at least 50 L/S at a pressure differential of 2.5 mm of water shall be provided for each dwelling unit."

2.4 LOCATION OF EQUIPMENT AND SYSTEM LAYOUT

The location of the heat exchanger equipment, and the system layout, depends as much on the design of the individual house and its heating system, as it does on the type of heat exchanger chosen.

Mechanical ventilation systems using air-to-air heat exchangers can be installed in a number of different ways. Most heat exchangers come as part of a package and may connect to existing ductwork in any part of the heated space, but most conveniently in a basement and close to an exterior wall. Some self-contained units can be window or wall mounted similar to a packaged air conditioner, and require no ductwork.

For houses with a forced-air heating system, the heat exchanger can be ducted into the system. Alternatively, the conditioned outside air can be supplied from the heat exchanger to the vicinity of main return air grille as shown in Figure 7. Most suppliers recommed the latter, since when directly connected, the cylic operation of the heating system fan upsets the pressure balance and therefore airflow in the heat exchanger.

For houses with or without a forced-air system, the heat exchanger unit can be connected to the exhaust ductwork from the kitchen and bathrooms.

Window mounted models are inexpensive and easy to install, but may be ineffective in achieving a satisfactory mixing of air.

As discussed earlier, of the five basic types of heat exchangers being considered in this report, namely, heat pipes, plate heat exchangers, heat wheels, coil loops and thermosiphons, the first three require that the two air streams come together in order to exchange heat. The coil loop and thermosiphon loop do not have this restriction, since the heat is transferred through piping and they can therefore be added to existing systems which have supply and exhaust air streams distant from each other.

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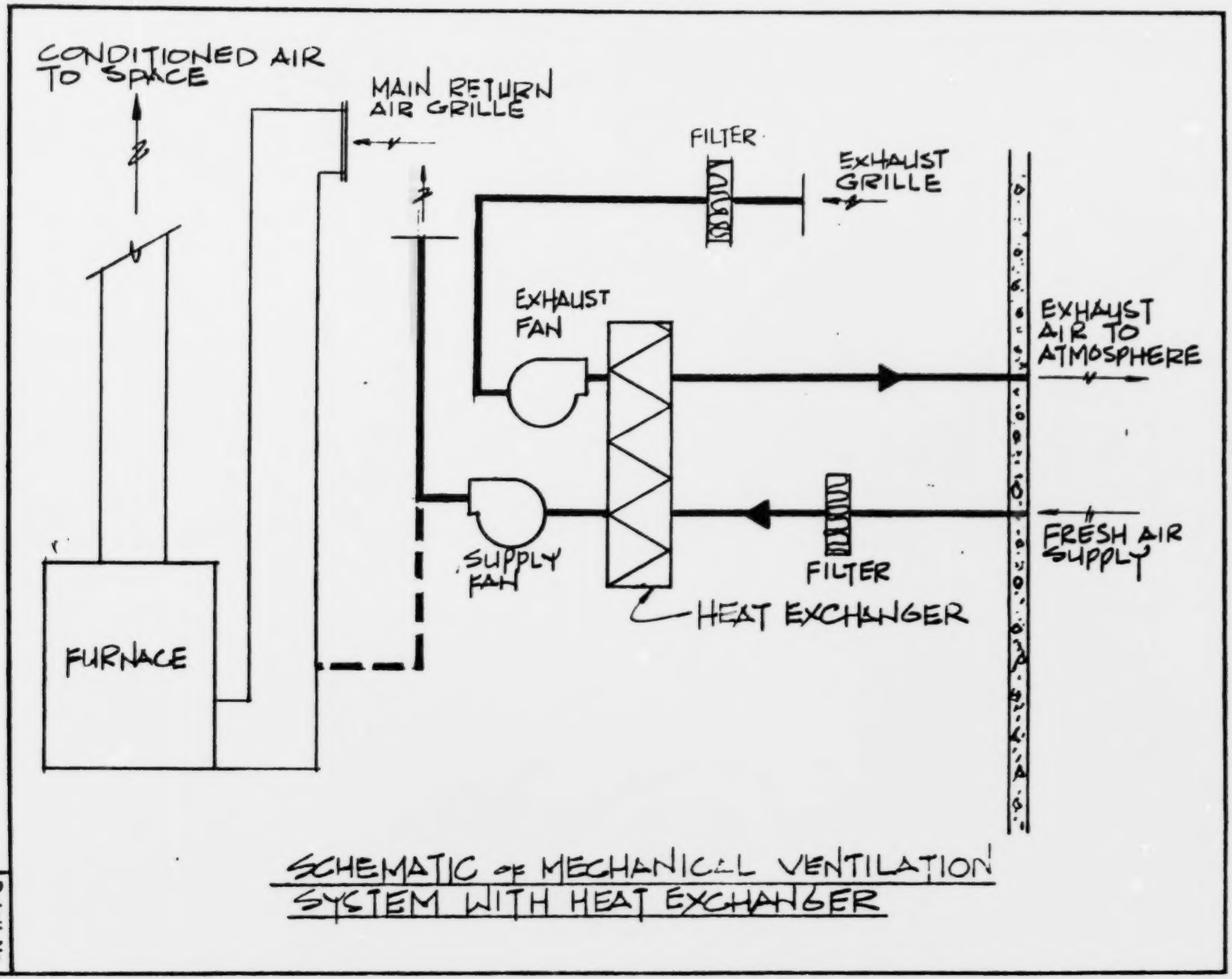


SCHEMATIC OF MECHANICAL VENTILATION SYSTEM WITH HEAT EXCHANGER

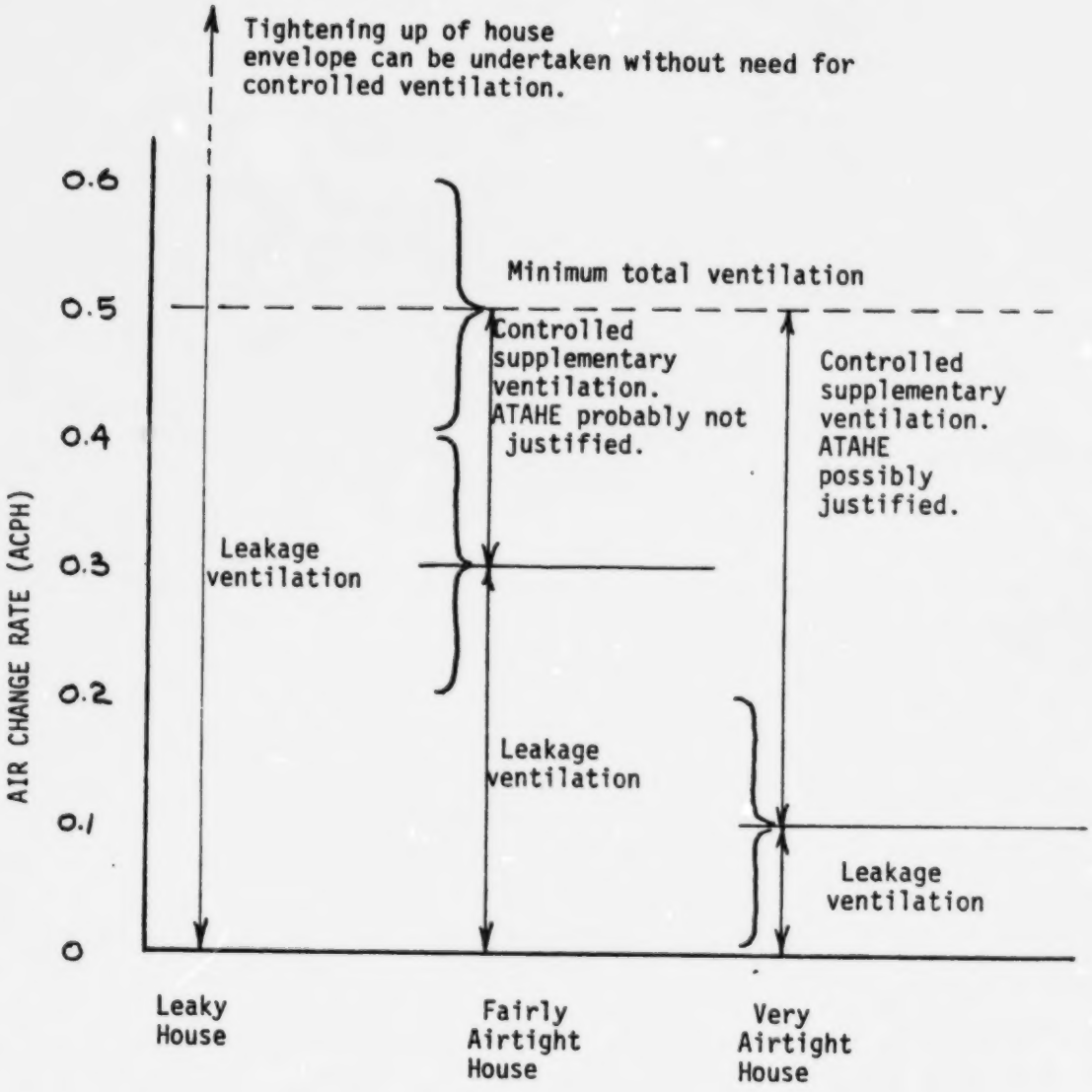
FIGURE 7 Detail No.

There is only benefit to be gained from an ATAHE if there is need for the ventilation it provides. The natural ventilation conditions required to justify an ATAHE are best illustrated diagramitically. At what point of airtightness it is justifiable to install an ATAHE, rather than a controlled supplementary ventilation system, is dependent on the several factors discussed in Section 4.0. Figure 8 illustrates the respective requirements for controlled supplementary ventilation for different degrees of house airtightness.

There are many ways to design a mechanical ventilation system incorporating an ATAHE. The best method for any given house depends on the design of the house and its heating system, as well as the requirements dictated by the number and habits of residents.

2.5 FROSTING AND ICING PROTECTION

One of the problems with ATAHE performance and reliability stems from the adverse conditions under which they are required to operate. Once freezing temperatures are reached in a heat exchanger, all types of ATAHEs may frost up in the exhaust side. Thus, some form of frost handling may be necessary.



Comparison of requirements for controlled supplementary ventilation.

Fig. 8

Frost protection may be achieved by:

1. Interrupting the incoming air flow;
2. Preheating the incoming air;
3. Reducing the energy transfer of the heat exchanger;
4. Reversing the air flow.

The easiest and lest expensive method of preventing frosting is by interrupting or reducing the incoming air flow. This results in the heat exchanger utilization being reduced for periods when it could be most effective.

Some of the commercially available ATAHE units are equipped with electric or liquid carrying preheat coils which warm the incoming air when its temperature may cause icing.

Prevention of frosting can also be achieved by reducing the energy transfer rate of the heat exchanger. This can be accomplished by bypassing a portion of the incoming air. This method can be used for all types of ATAHE's. Reduction of the supply flow rate from balanced flow to 50% of the exhaust rate can reduce the frost threshold by as much as 11°C (20°F). This is discussed in more detail in Section 5.

Some of the heat exchangers considered in this report can employ other methods of reducing their energy transfer:

- Heat pipes and thermosiphons can be tilted, thus controlling their ability to transfer heat;
- Heat wheels can have their energy effectiveness reduced by slowing their rotation.

Apart from the problem of frosting which may occur on the heat transfer surfaces of heat exchangers, there is a danger that improper drainage of the condensate may cause icing within the heat exchanger of ductwork. Therefore, care must be taken when designing and installing the heat exchanger to see that any condensate or water produced by defrosting be drained away from the cold end and removed from the heat exchanger.

2.6 THERMAL, MECHANICAL, ELECTRICAL AND ACOUSTIC CRITERIA

The thermal performance of an ATAHE unit to transfer sensible, latent, or total heat from one air stream to the other has been discussed in some detail. Besides thermal requirements, mechanical, electrical and acoustic criteria need also be investigated.

The size and weight of a heat exchanging unit is an important factor when installation is being considered. A convenient size for such units would be one which could fit between joists, typically 360 mm (14 inches) apart and thus be out of the way. A domestic heat exchanger of this size should weigh no more than 45 kg (100 lbs.) thus allowing one man to handle any cleaning or maintenance if necessary.

The heat exchanger unit should require as little energy as possible to operate it, since energy saved by heating load reduction will be decreased by operating energy expenditure. Exchangers with plates, pipes and thermosiphons require no power other than to run the fans needed to produce the ventilation. Wheels and coil loops require an extra power input to drive their heat transfer medium as well as the fans for ventilation. Referring to Figure 7, it should be noted that by placing the fans as shown, heat given off by the supply and exhaust fans is partially recovered, increasing the system effectiveness.

Installation of a simple alarm may be considered in the form of a pilot light to indicate malfunction of the ATAHE unit. Such devices are easy to produce and relatively inexpensive, yet would increase the reliability of the system. The protection of the electrical system from moisture, which may be produced by frosting and condensation, should also be considered.

In some instances, it may be beneficial to run the ATAHE for only a certain portion of the year. In this case, the owner should be provided with a means of manually turning the heat exchanger on and off independently of the mechanical ventilation system.

Acoustics also may require consideration. Since most of the ATAHE units are located indoors, it is not expected that the outdoor noise levels would pose a problem (most municipal bylaws allow a noise level of 55dB at the property line [28]). The acceptable noise level for homes and bedrooms is found to be 35 dB (29), and heat exchanger/ventilation systems should be designed with this in mind.

2.7 REQUIREMENTS FOR DIFFERENT REGIONS OF CANADA

Optimization of design for an ATAHE is dependent on the conditions in which it will have to operate, as well as the energy economics involved. It is probable that as the market grows, units will be developed to correspond to particular demands. The primary influencing factors that will vary on a regional basis are design temperature and interior humidity. The former varies for example from -7°C in Vancouver to -39° in Churchill Humidity levels are governed by two influences, the amount of moisture entering the interior air from household activities and people, and the moisture level a house can tolerate before damaging condensation will occur. Certainly household,

especially cooking, habits differ regionally, with the practice in some rural and northern areas of maintaining a pot on the boil.

The combined effects of outside temperature and interior humidity levels will govern the extent of defrosting required. Further, high electricity costs and/or accessibility for servicing may influence the type and complexity of the defrosting method used.

2.8 OPERATION CHARACTERISTICS

A copy of operation characteristics for various types of ATAHE units as reported by manufacturers and independent testing facilities, can be found in Appendix II. The results, in most cases, compare favourably with the performance requirements outlined in this report, and verify the applicability of the design parameters.

3.0 OPERATION OF ATAHEs AND FROST PROTECTION EFFECTIVENESS

3.1 FROST PROTECTION - OPERATION AND EFFECTIVENESS

In Section 2 we discussed the freezing problem encountered by all types of Air-to-Air Heat Exchangers and some of the possible methods of frost prevention.

In actual practice, the various types of ATAHE's employ a number of different frost prevention methods.

Some of the commercially available heat exchanger units, (Flakt, Mitsubishi), come equipped with electrical preheat coils to heat the incoming fresh air to temperatures above the freezing point.

The heat exchanger unit designed and built by a Canadian firm (Air Change Co. of Toronto), senses freezing conditions using a temperature sensor on the warm air side of the fresh air stream and reduces its effectiveness by shutting off the supply air fan while these conditions exist.

The rotary wheel built by Ontario Hydro has the ability to reduce its effectiveness by a reduction in its rotation speed, should frosting occur. The extent of frosting was observed by monitoring the differential pressure across the heat exchanger. The onset of frosting would cause an increase in this pressure. However, in a recent field test of the heat exchanger in one of the HUDAC Mark XI homes (27), the operation was found to be free of any frost formation which would have influenced the operating effectiveness.

Other studies (1,16,23,33,34,35) have shown that proper design of the heat exchanger system including preheat sections, and control of indoor humidity, should eliminate the frosting problem in all but the most severe instances.

3.2 TROUBLE FREE OPERATION

In several tests of ATAHE performance in experimental testing facilities (35,36) and installed in residential houses (23,27), there have been no major problems encountered during heat exchanger operation.

Most preliminary test reports are in agreement that the heat exchangers are fulfilling their primary purposes of reducing the heating load due to ventilation and controlling the build-up of indoor humidity and toxic gases. However, there are problems, especially with installation practices, leaving much scope for improvement.

A problem has arisen with several installations in the prairie region where inadequate provision has been made for defrosting. The build-up of ice in the exhaust air stream throttles the flow, resulting in a degree of pressurization of the house due to the continuing operation of the supply fan. The pressurization then causes exfiltration which can be so detrimental to the house envelope.

There has been some mention that the fans in a particular type of heat exchanger, produced an annoying noise (37), but this appears to be an isolated incident.

Development of ATAHEs in Canada has been directed exclusively for winter application, since there is no real need for heat exchanger during summer.

3.3 MAINTENANCE

The ATAHE units should be designed to require as little maintenance as possible. Simple operations to ensure continued effectiveness, such as cleaning of the air filters, which can be performed by the homeowner, should be all that is required.

The units should be installed in a location easily and readily accessible. Space around the unit should be allowed for easy removal. Filters, whether they are part of the unit or duct installed, should also be easily accessible for removal and change.

The homeowner should be provided with a clearly understandable instruction manual explaining the function of the unit, its operation and maintenance required.

3.4 INTERACTION WITH OTHER SYSTEMS

One of the problems expected as a result of the operation of a mechanical ventilation Air-to-Air Heat

Exchanger system of a house, is the effect of fluctuating static pressure conditions during the operation of a combustion furnace.

Fluctuations may arise due to any of three reasons:

1. The unit may be turned on and off automatically by means of a humidity controller or manually;
2. If frosting occurs, the exhaust will gradually be throttled until the defrosting cycle commences. The greater the throttling, the greater the pressure rise created in the house.
3. If defrosting is achieved by stopping the intake, some depressurization will occur, gradually increasing until completion of the cycle.

These changes may upset the operation of combustion devices in a house. Care must therefore be exercised in the installation of an ATAHE.

This problem may be avoided by enclosing the furnace area and providing make-up air for the burner directly from outside. However another potential problem presents itself, that of over-cooling the furnace and flue. Oversizing of a furnace may occur even more frequently in energy efficient housing than in others. As a consequence, even in cold weather, there may be long periods between the

furnace operation; and when heat is required, the furance operates only for a short while. If the furance area is vented from outside, the net effect can be an over chilling of the furnace, with consequent loss in combustion efficiency and little opportunity for the flue or chimney to warm up. In the worst case condensation freezes in a flue and can cause a blockage.

The remedy is to ensure that the furnace area not be allowed to become too cold, a minimum of 10°C is suggested by EMR.

4.0 COST EFFECTIVENESS OF ATAHE SYSTEMS

In order to determine the cost effectiveness of ATAHE systems, consideration must be given to both capital cost of equipment and operating cost (9). In addition, the reduction in energy consumption should be evaluated.

When evaluating capital costs, the following cost items should be considered:

1) Cost of ATAHE unit;
2) Installation of ATAHE units;
3) Additional ductwork and/or piping;
4) Cost of larger and additional fans to overcome pressure drops through ATAHE and cost of additional pumps;
5) Additional air filtration;
6) Capacity controls;
7) Auxiliary heaters for frost control;
8) Savings in heating equipment and associated pipes, pumps, and electrical equipment, if any, due to reduced ventilation load.

Similarly, when calculating the operating cost of ATAHE units, the following items should be evaluated:

1) Cost of energy to operate additional and larger fans, pumps, defrost heaters, and unit drives;
2) Maintenance cost of ATAHE and associated fans, ducts, pumps and pipes;
3) Maintenance of additional filtration;
4) Savings of annual heating energy.

4.1 METHODOLOGY

As reflected in this study, there are numerous types of ATAHE units, there are many different ways to install them and many different ways to control them. In order to carry out a meaningful economic analysis, it is proposed to reduce the number of variables to a manageable quantity by establishing unit prices for both capital and operating costs.

These unit prices are intended to be used as a guide only, and it is up to the designers to adjust these cost

figures as more experience is gained in this newly developing market. At the present state of the art, most unit costs are assumptions and are open to discussion.

Also, a great number of procedures of varying complexity are available to evaluate the financial viability of projects. Payback period, return on investment, break-even period, benefit/cost analysis, and life cycle costing are various methods in use today.

The most straight forward and most understandable of these methods is the simple payback period, which is the time period during which the accumulated savings are equal to the initial capital cost. This method is considered as the first level financial analysis, used to screen out projects with low return on investment, before the application of a more sophisticated examination.

In the following economic analysis, all calculations are arranged so that they can be adjusted easily for any particular application (42).

4.2 CAPITAL COST

Since the manufacturing industry is in its infancy, the actual performance of mass produced units is uncertain, as is the capital cost of installations. Thus it is proposed to base calculations on two extreme values, one on high capital cost, and one on low capital cost.

The cost of the basic ATAHE unit was discussed in Section 1. As shown in Table 1, respondents to the survey indicated that the average cost of basic unit, is $800,00. Some manufacturers have suggested that as the market expands the initial cost could decrease to as low as $200.00 to $400.00 for production of 5,000 units.

The capital cost for additional ductwork, piping, fans, pumps, preheat, controls, etc. and installaion, which are required for an ATAHE system over and above a standard santitary and kitchen exhaust system may be estimated at $200.00.

The high capital cost will be assumed at $800.00 (unit cost of $600.00 plus $200.00 for installation and auxiliary equipment) and the low cost will be $400.00 (unit cost at $200.00). For units without electric frost protection, the capital cost is assumed to be less by $50.00.

4.3 OPERATING COSTS

For calculation of operating costs, assumptions and unit prices are shown in Table 6.

Energy costs are based on 1982 prices in Toronto.

Assumptions are based on the following reasoning:

1. A sensible heat recovery efficiency of 70% is calculated, which has to be adjusted for any specific ATAHE.

2. Fan power requirements are based on supply fan only, since the heat exchanger exhaust fan replaces one which otherwise would have to be installed to control ventilation.

TABLE 6 ASSUMPTIONS AND UNIT PRICES FOR ECONOMIC ANALYSIS

Ventilation Rate	Q	50 L/s (106 cfm)
Location		Toronto, Ontario
Annual Degree-Days	D	4082 DD (7380 °F DD)
Outside Design Temperature	TO	-18°C
Inside Design Temperature	TI	21°C
Sensible Heat Recovery Efficiency	E	70%
Supply Fan Power	F	50W
Fan Operating Hours	HF	4640 Hrs/Yr
Preheat Power	P	1000W
Frost Formation Hours	HP	700 Hrs/Yr

Energy Prices (Toronto January 1, 1982 Rates)	$ MMB	$ GJ	$ MWH
Electricity 3.68 cents/Kw.h	11.00	10.40	36.8
Gas 4.80 $/MCF (@ 70% Seasonal Eff.)	6.90	6.50	23.1
Oil No. 2 1.20 $/IG (@ 65% Seasonal Eff.)	11.00	10.40	36.8

Annual Maintenance and Filtration Cost $20.00

3. Fans are in operation whenever the outdoor temperature is below 12°C (54°F), or 4640 hours in Toronto (Figure 9). See para 6 below for rationale for using 12°C.

4. With the assumption of a 70% sensible heat recovery we can estimate the outdoor temperature at which frosting will commence.

From Section 2.2 we know that:

$$\mathcal{E}_{sh} = \frac{T_3 - T_4}{T_3 - T_1} \quad \text{or} \quad \frac{T_1 - T_4}{T_1 - T_0}$$

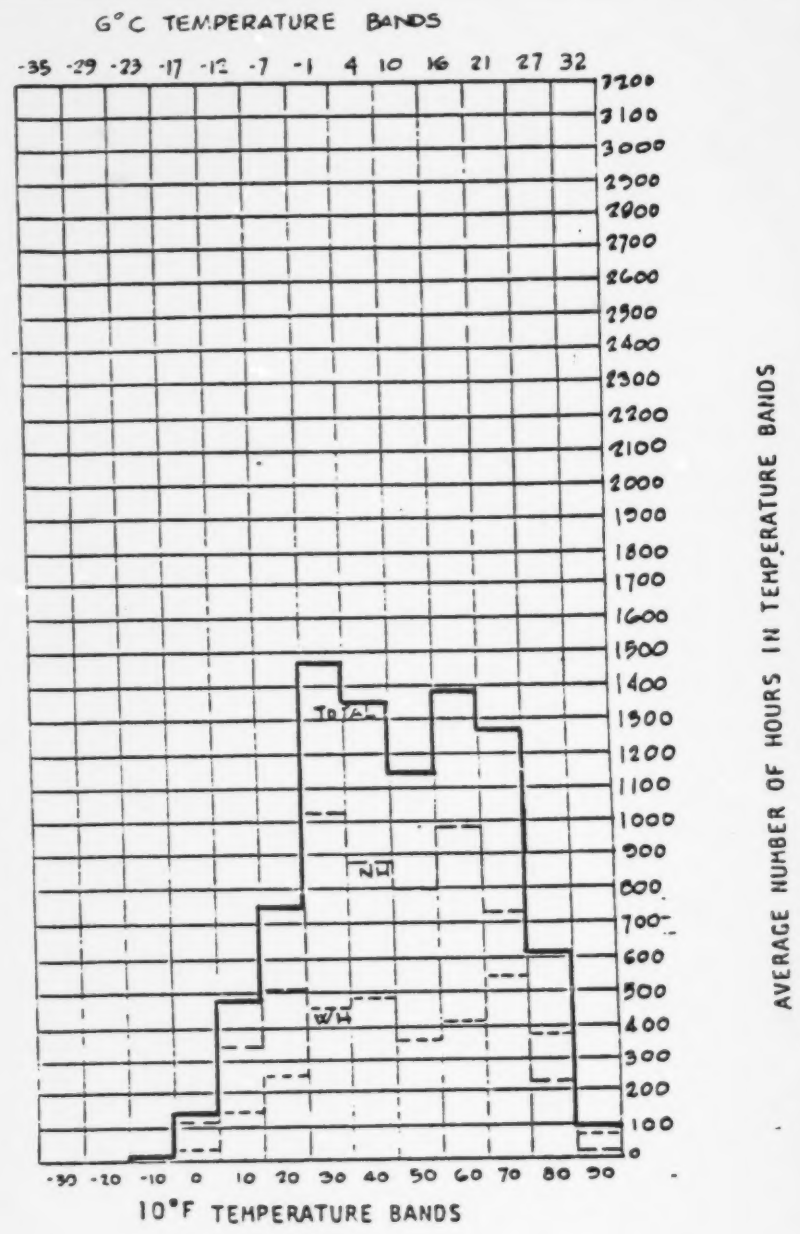
where T_4 is the temperature of the cooled exhaust.

If T_4 is assumed to be 0°C

Then $\mathcal{E}_{sh} = 0.7 = \frac{21° - 0°}{21° - T_0}$

from which $T_0 = -9°C$

From Figure 9 we can determine that in Toronto the temperature is at or below this level for 700 hours per year.



TORONTO

WORKING HOURS (WH)	0800-1900	3131
NON WORKING HOURS (NH)		
	2000-0700 and WEEKENDS	5634
TOTAL		8765

FREQUENCY DISTRIBUTION OF HOURLY TEMPERATURES

Standard Number

Standard Name

Drawn :
Date :
Checked :
Revised :

Detail No.

FIGURE 9

5. Annual maintenance and filtration costs account for one change of filters per year and regular lubrication.

6. Heat loads have been typically determined on the basis of the temperature differential between the house interior and the design temperature. This is satisfactory so long as the Internal Heat Gains, i.e. arising from occupants, cooking, hot water, solar, etc., are small compared with the total heating requirement.

   Since, as is explained elsewhere, ATAHE's are viable only with relatively airtight, and therefore by implication energy efficient housing, the internal heat gain becomes a significant proportion of the total heat required. For the sake of example we will assume here that the internal heat gain will maintain an interior temperature T_1 of 21°C until the external temperature T_0 drops below 12°C. Above this outdoor temperature ventilation is called for to stop the internal temperature exceeding 21°C.

   The temperature at which the heat losses from the house equal the internal gains is called the break even temperature T_B. An ATAHE only makes a

contribution to heat conservation below the break even temperature. For Toronto, and a T_B of 12°C the hours below this temperature are 4640 hours per year.

7. Preheat is provided by electric heating coil. The frost protection heater adds heat to the incoming outdoor air. The process of adding heat, however, reduces the temperature difference between exhaust and outdoor airstreams, thus reduces the overall recovery efficiency. It is assumed that 70% of frost preheat is useful.

8. A shut-down system without preheat would be off for 700 hours each year to prevent frosting. During this period the outdoor air infiltrating to replace the exhaust air would have to be heated without heat recovery.

9. A method of calculation is shown which prorates degree days below a given temperature according to the number of hours per year below that temperature. Thus knowing the degree days below 18°C for Toronto and having the frequency distribution of hourly temperature (Figure 12) the equivalent degree days may be computed. Alternatively, if degree day tables are available for the appropriate location then these may be used. Sample operating cost calculations are carried out in Tables 7 and 8.

4.4 CALCULATION OF PAYBACK PERIOD

Sample calculations are carried out to estimate sample payback periods for both preheat and shut-down ATHAE systems in Tables 9 and 10.

Based on the assumptions stated, the calculated payback periods vary between 5 to 22 years for the Toronto area and similar climatic regions.

The calculations illustrated here can be repeated for any ATAHE system with different efficiencies, fuel costs and all other parameters to determine the economic viability of heat recovery systems.

TABLE 7

SAMPLE OPERATING COST CALCULATIONS - with preheat

Ventilation Heat Loss Without Recovery (42) $H = 1.23 \times Q \times (TI - TO) = 1.23 \times 50 \times (21 + 18) = 2.4$ kW
Annual Heating Energy Without Recovery $EH = \frac{H}{TI - TO} \times D \times \frac{\text{hours below } T_B}{\text{hours below } T_I} \times 24$ $= \frac{2.4}{39} \times 4082 \times \frac{4640}{6970} \times 24 = 4014$ kW.h = 14.4 GJ
Annual Fan Energy $EF = F \times HF = 50$ W x 4640 Hrs = 232 kW.h
Annual Preheat Energy $EP = P \times HP = 1$ kW x 700 Hrs = 700 kW.h

TABLE 8

SAMPLE OPERATING COST CALCULATIONS - with shut down

Ventilation Heat Loss Without Recovery
(same as in Table 7) $H = 2.4$ KW

Annual Heat Recovery Without Recovery
(Same as in Table 7) $E_H = 4014$ KWh

Heating Required between -9°C and +12°C

$$\frac{H}{T_1 - T_0} \times D \times \frac{\text{hours between -9°C and +12°C}}{\text{Total below +18°C}} \times 24$$

$$\frac{2.4}{39} \times 4082 \times \frac{(4640 - 700)}{6970} \times 24$$

$$= 3404 \text{ KWH}$$

of this (1 - 70%) is required = 1021 KWH

Annual Fan Energy

$$EF = F \times HF = 50W \times (4640 - 700)$$

$$= 197 \text{ KWH}$$

Heating Required Below -9°C

$$\frac{2.4}{39} \times 4082 \times \frac{705}{6909} \times 24$$

$$= 609 \text{ KWH}$$

of this all is required, since no heat exchange takes place below -9°C

TABLE 9 PAYBACK PERIOD

ATAHE SYSTEM WITH PREHEAT

	ENERGY	COST - $/YR		
	kW.h	ELEC	GAS	OIL
A. WITHOUT HEAT RECOVERY				
Heating Required	4014	148	93	148
B. WITH HEAT RECOVERY				
Heating required				
(0.3 x 4014)	1204			
Less Fan Energy (1.0 Useful)	-232	8	8	8
Less Preheat Energy (700 x .7 useful) (0.7 Useful)	-490	18	18	18
Additional Energy Required to heat ventilation air.	476	18	11	18
Maintenance		20	20	20
Net Cost		64	57	64
C. SAVING A - B		84	36	84
SIMPLE PAYBACK PERIOD-YEARS				
Capital Cost @ $400. / C		4.8	11.1	4.8
@ $800. / C		9.6	22.2	9.6

TABLE 10: PAYBACK PERIOD

ATAHE SYSTEM WITH SHUT-DOWN

	ENERGY	COST - $/YR		
	KW.h	ELEC	GAS	OIL
A. WITHOUT HEAT RECOVERY				
Heating Required	4014	148	93	148
B. WITH HEAT RECOVERY				
Heating Required Below -9°C (0.3 x 3404)	1021			
Heating Required Below -7°C	609			
Less Fan Energy 3940 H (1.0 Useful)	197	7	7	7
Net Energy Required	1433	53	33	53
Maintenance		20	20	20
Net Cost		80	60	80
C. SAVING A - B		68	33	68
SIMPLE PAYBACK PERIOD - YEARS				
Capital Cost @ $350. / C		5.1	10.6	5.1
@ $750. / C		11.0	22	11.0

5.0 USE OF ATAHE's TO MAINTAIN A NEGATIVE PRESSURE WITHIN THE HOUSE ENVELOPE

In houses where humidity levels are relatively high during the wintertime, exfiltration of warm humid air through cracks and openings in the house envelope may lead to condensation when this air flow encounters cold surfaces. This can result in the reduction in thermal insulating properties of materials and deterioration of structural elements in the house due to wetting.

This potential problem would be reduced if air pressure inside the house was lower than the pressure outside, so that moderate infiltration occurred, rather than exfiltration. Operating an ATAHE so that the exhaust flow exceeded the supply by an appropriate amount would be one way to establish the necessary pressure difference.

In order to estimate the required difference between the exhaust and supply, it is necessary to know the existing air pressure distribution and air leakage characteristics of the house.

In cold weather, when the outside air temperature is much lower than the inside temperature, a pressure differential between the inside and outside of the house is created due to the difference in densities of the air. This so called stack, or chimney effect, results in infiltration of the air at lower levels of the house and exfiltration from the upper levels. The line on the vertical surface of the house where the pressure difference across the walls is zero, and no infiltration or exfiltration occurs, is called the neutral pressure plane, (NPP) or neutral plane Figure 10 shows an illustration of the pressure variations and the resulting air flow patterns.

The height of the neutral plane depends on certain characteristics of the house. For a house with no chimney and a fairly even distribution of openings, the neutral plane is close to the middle level of the house. An open chimney acts as an opening at the top of the house and raises the neutral plane. For this reason, a house with a chimney would have less air leakage out through the ceiling and consequently less potential for condensation.

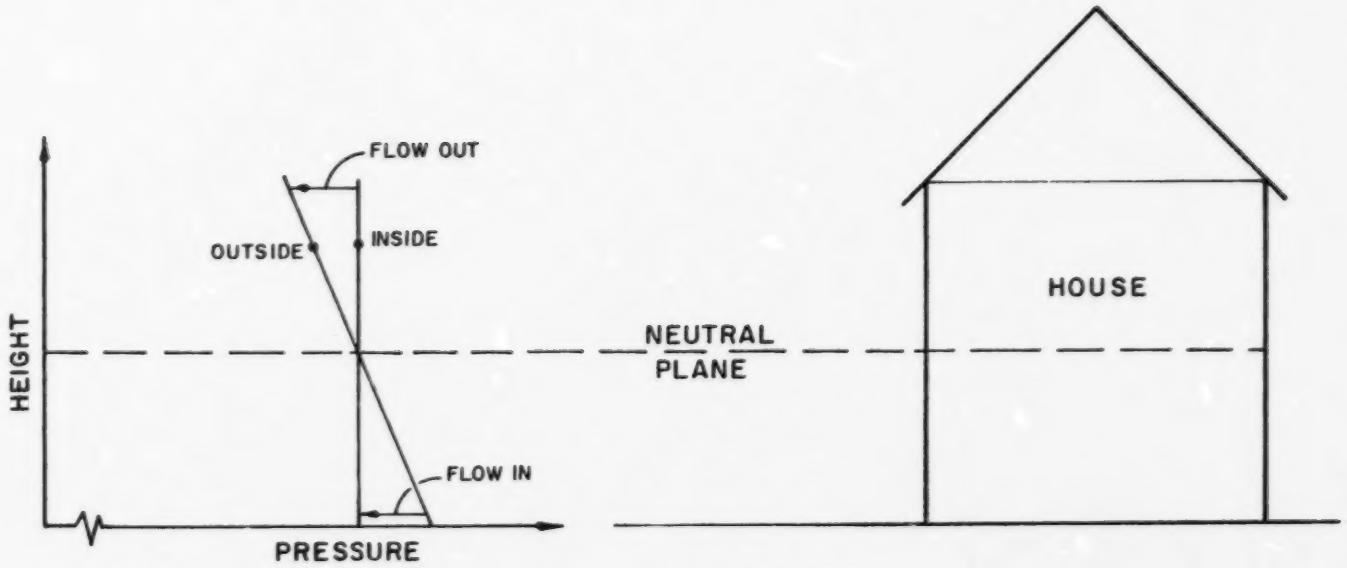


FIG. 10 STACK EFFECT WITH NEUTRAL PLANE AT MID HEIGHT

In order to eliminate exfiltration due to the stack effect, the neutral plane would have to be raised to the highest ceiling level of the house, as illustrated in Figure 14.

For this to happen, all of the air infiltrating into the house would have to be removed by the ATAHE, or by other exhaust equipment. For a house with no chimney and no other exhausts, the excess amount of air removed by the ATAHE would be the total infiltration into the house.

If the ventilation air requirement for the house is 50 L/s as suggested in section 2.3, then the fresh air supplied through the ATAHE would be 50 L/s minus the air infiltration into the house.

For a house with a volume of 400 cubic meters, which is representative of the HUDAC Mark XI houses in Ottawa,[44] 50 L/s would represent an air change rate of 0.45 air changes per hour.

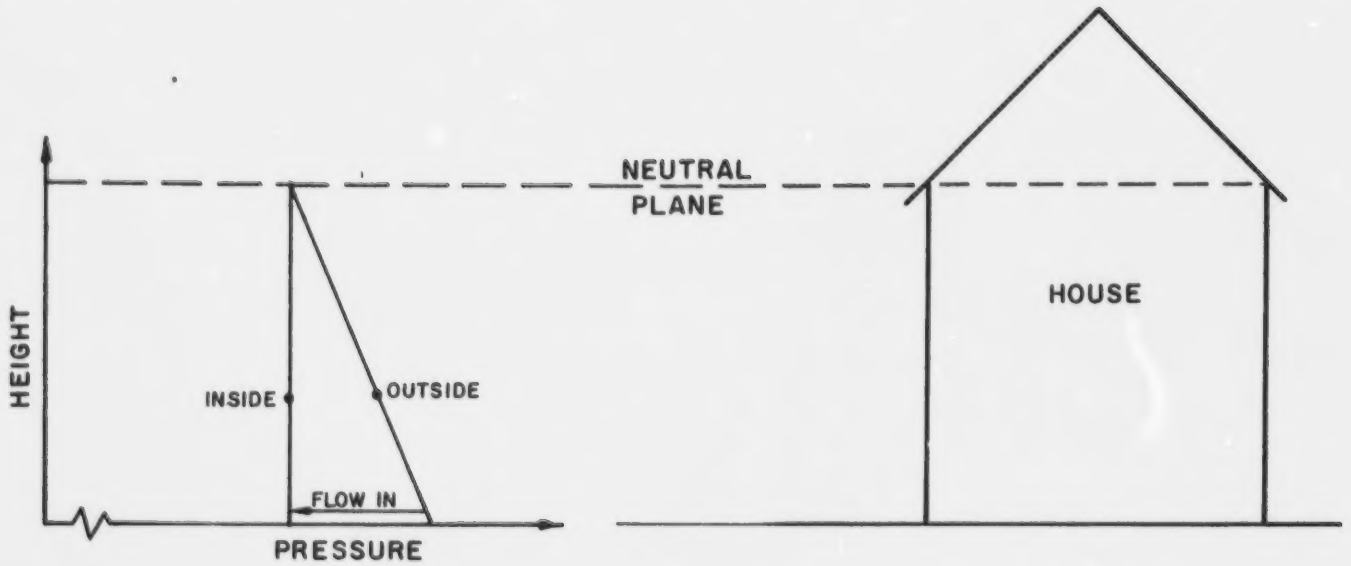


FIG. 11 PRESSURE DIFFERENTIAL WITH NEUTRAL PLANE AT CEILING HEIGHT

The air leakage rate for a house can be expressed in terms of pressure differentials by the equation

$$Q = CA\,(\Delta P)^n$$

where

Q = air leakage rate, L/s

C = flow coefficient, $L/s \cdot m^2\ (Pa)^n$

A = area of building envelope, m^2

Δp = pressure difference across building envelope Pa

n = flow exponent

For the HUDAC Mark XI homes, the value of n is 0.71 and C is 0.075 for the H4 house.

The pressure difference, Δp from the stack effect can be found using Equation 2 from Chapter 21 of the ASHRAE Handbook of Fundamentals.

$$Pc = 0.0342\ PH\ \left(\frac{1}{T_o} - \frac{1}{T_i}\right)$$

Where

P_c = theoretical pressure difference across enclosure due to chimney effect, Pa

P = atmospheric pressure, Pa

H = distance to neutral pressure level, m

T_o = absolute temperature outside, ^oK

T_i = absolute temperature inside, ^oK

If we assume an inside temperature of 21^{o}C and an outside temperature of -19^{o}C, a height of the neutral plane of 2.5 meters, or about one half on the height of the house, and an atmospheric pressure of 1×10^5 Pa, then the pressure difference across the wall is

$$\Delta p = 0.0342 \times 10^5 \times 2.5 \left(1/254 - 1/294\right)$$

$$\Delta p = 4.6 \text{ Pa}$$

If we assume this pressure difference in the equation for air leakage and assume that most of the openings are located near the upper and lower levels of the house, the resultant leakage rate is

$$Q = 0.075 \times \frac{227.7 \text{ m}^2}{2} (4.6)^{.71}$$

$$Q = 25 \text{ L/sec.}$$

where $\frac{227.7}{2}$ m^2 is one half of the area of envelope of the house, since infiltration occurs through one half of the envelope, and exfiltration occurs through the other half. This approximation will tend to over estimate the infiltration since not all openings will be subjected to the maximum pressure difference. Shaw[44] has measured Q as 23 L/sec under actual conditions.

If the neutral height is now raised to the top of the house, i.e. 5 meters, the pressure difference becomes

$$\Delta p = 0.0342 \times 10^5 \times 5 \left(\frac{1}{254} - \frac{1}{294}\right)$$

$$\Delta p = 9.3 \text{ Pa}$$

This gives an air leakage rate of

$$Q = 0.075 \times \frac{227.7}{2} \ m^2 \ (9.3)^{.71}$$

$$Q = 42 \text{ L/sec}$$

This approximation assumes that only the lower openings contribute to air infiltration and that they are subject to

the maximum pressure difference.

If the new leakage rate of 42 L/sec is compared to the required ventilation rate of 50 L/sec it can be seen that the supply air stream of an ATAHE would be reduced to close to zero under this condition. Thus the ATAHE would function as an exhaust fan only.

This example shows that at the extreme design conditions the air leakage caused by effects approaches the design ventilation rate of 50 L/S. Wind pressures add further to the air change rate. AT temperature differentials the leakage rate will be correspondingly lower and differences between inlet and exhaust airflow rates can be accordingly less to control the elevation of the NPP.

But as Figure 9 helps illustrate, the average heating season temperature is very much higher than the design temperature. For Toronto the average heating season temperature is +2°C. At this temperature Q works out to be 26 L/S.

Under this condition to achieve the design ventilation rate the exhaust through the heat exchanger should be 50 L/S and flow through the intake (50 - 26) = 24 L/S. At this difference in air flows the effectiveness of the ATAHE will certainly be reduced.

In recent tests on a number of installed ATAHE systems it was found that the rate of supply air exceeded the exhaust; the reverse of what is being advocated above. This is generally attributable to there being a more restrictive exhaust collection ducting system than supply system. This points to the possible need for heat exchangers to have a more powerful exhaust fan then supply fan, not only to match the ducting back pressure, but also to contribute to producing the requisite negative pressure. A further benefit which may be traded-off against the decreased effectiveness, is that it extends the frost free operation time, since it amounts to the third procedure which may be used for defrosting, described in Section 2.5.

The reader should be reminded again though of the concerns discussed in Section 3.4, namely of depressurizing a home that has naturally apirated combustion equipment. Excessive depressurization can reverse air flow in flues, with the possibility of combustion products entering the house. This must be avoided at all costs. Either there is no place for an ATAHE in such housing, or interlocking controls or other precautions must be taken to ensure that the depressurization cannot occur during operation of the combustion equipment.

In summary it appears that ATAHE's in some circumstances and if properly installed and controlled could be used to reduce air pressure in a house. What is clear however is that it is not a simple matter, and that natural ventilation rates and ventilation requirements of any combustion systems must be understood before installing an ATAHE.

6.0 TEST FACILITY REQUIREMENTS

The following section describes the sort of equipment required for carrying out testing of simulated service conditions. They are not intended to be explicit but to serve as the basis for a test facility specification.

6.1 GENERAL DESCRIPTION

Design flow rates for the test facility of single dwelling residential ATAHEs are 25 L/s (50 cfm) minimum to 190 L/s (400 cfm) maximum.[1,46] The test facility should include all necessary provisions for:[47]

(a) Heating, cooling, humidifying, and/or dehumidifying the air to simulate the following field conditions:

Fresh Air Intake	-40°C to 16°C (-40°F to 60°F) for winter conditions and 29°C to 38°C (85°F to 100°F) for optional summer conditions with relative humidity = ambient in both cases
Exhaust Air	+18°C to +24°C (+65°F to 75°F) with a relative humidty range of 20 - 70%
Ambient Air	+10°C to 36°C (=50°F to +97°F) with relative humidity = to ambient. Where ambient air is the air surrounding the ATAHE during tests.

(b) Measurements of exhaust and supply air flow rate both upstream and downstream of the ATAHE with provisions for establishing larger flow rates of exhaust air than supply air.

(c) Varying the pressure differential between the supply duct and the exhaust duct.

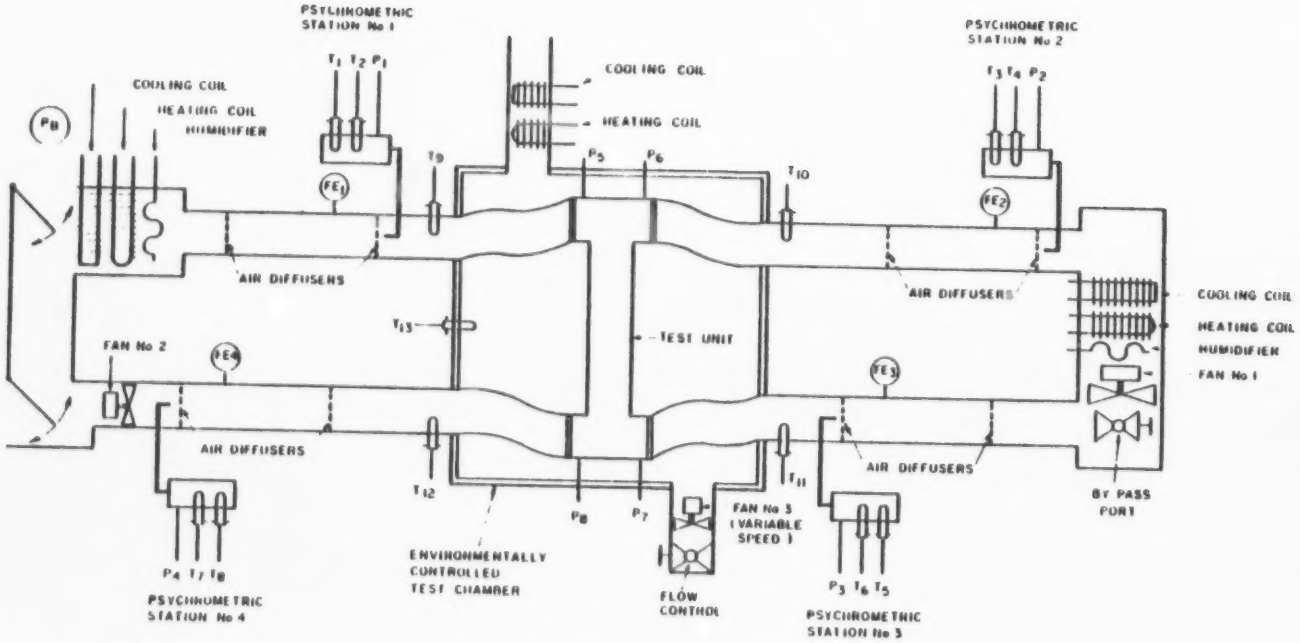
(d) Measurements of static pressure, dry-bulb and wet-bulb temperatures of exhaust and supply air both upstream and downstream of the ATAHE.

(e) Injection and measurement of tracer gases.

(f) Measurement of noise levels of the ATAHE.

(g) Measurement of electrical power consumption of the ATAHE and associated equipment.

A schematic of a typical test facility is shown in Figure 12. The facility should be constructed for ease of interchangeability of the air to air test heat exchangers. As



LEGEND

FE_1	FLOW ELEMENT	SUPPLY INLET	P_4	PRESSURE	PSYCHROMETRIC STATION No 4	T_5	WET BULB TEMPERATURE STATION No 2
FE_2	" "	SUPPLY OUTLET	P_5	PRESSURE	SUPPLY INLET	T_4	DRY BULB " " No 2
FE_3	" "	EXHAUST INLET	P_6	"	SUPPLY OUTLET	T_5	WET BULB " " No 3
FE_4	" "	EXHAUST OUTLET	P_7	"	EXHAUST INLET	T_6	DRY BULB " " No 3
P_1	PRESSURE	PSYCHROMETRIC STATION No 1	P_8	"	EXHAUST OUTLET	T_7	WET BULB " " No 4
P_2	"	" " No 2	T_1	WET BULB	TEMPERATURE STATION No 1	T_8	DRY BULB " " No 4
P_3	"	" " No 3	T_2	DRY BULB	" " No 1	T_9	SUPPLY INLET TEMPERATURE
						T_{10}	SUPPLY OUTLET TEMPERATURE
						T_{11}	EXHAUST INLET TEMPERATURE
						T_{12}	" OUTLET "
						T_{13}	TEST UNIT AMBIENT TEMPERATURE

FIGURE 12

SCHEMATIC OF TEST FACILITY FOR AIR TO AIR HEAT EXCHANGERS

indicated in Figure 12, outside air brought into the supply side of the loop is cooled or heated as prescribed by the test schedule (Section 8.2.1) and then humidified as required for ambient humidity conditions. Air leaving this treatment station enters one of the four combination air flow and psychrometric measuring stations then continues through the fresh air intake side of the ATAHE to the second air flow/psychrometric station where changes in air conditions and flow rates are measured. Air flow continues through the second treatment station where air conditions are adjusted as prescribed by the test schedule to simulate normal exhaust air. The first of the two fans, located after the second treatment station, induces air flow through the fresh air side of the test loop. This fan, in conjunction with the second, also induces air flow through the exhaust side of the test loop which enables adjustment of a differential pressure between the two air streams. Construction of the exhaust section of the test loop is essentially the same as the intake section with two air flow/psychrometric stations located before and after the ATAHE test unit and the second fan (variable speed) located downstream of the system to control air flow. The ATAHE test unit is enclosed in an insulated chamber with provision made for adjusting ambient conditions surrounding

the unit to simulate application environment of the ATAHE, i.e. attic, crawlspace, interior installation, etc.

6.2 TEMPERATURE MEASUREMENT

Temperature measurement and instrumentation unless otherwise noted shall be in accordance with ASHRAE Standard 41.1-74, Standard Measurement Guide: Section on Temperature Measurement.

Temperature measurement shall be made with one or more of the following instruments:

(a) Mercury in glass thermometers

(b) Thermocouples

(c) Electrical resistance thermometers

Accuracy shall be within the following limits:

(a) Wet and dry bulb air temperatures: $\pm$ 0.1°C

(b) Other dry bulb temperatures $\pm$ 0.3°C.

In no case shall the smallest scale division of the temperature measuring instrument exceed twice the specified accuracy. For example, for the specified accuracy of $\pm$ 0.1°C the smallest scale division shall not exceed 0.2°C.

When an accuracy closer than $\pm$0.3°C is specified, the instruments shall be calibrated by comparison with a thermometer which has a National Bureau of Standards certification in the range of use, or a thermometer with a National Bureau of Standards certification shall be used.

Whenever possible, temperature measuring instruments used to measure the change in temperature shall be arranged so that they can be readily interchanged between supply and exhaust positions to improve accuracy by utilizing average value.

6.3 PRESSURE MEASUREMENT

6.3.1 INSTRUMENTATION

Pressure measurements shall be made with liquid column manometers or pressure transducers. Accuracy of pressure measuring instruments shall permit measurements within $\pm$ 1% of the reading. In no case shall the smallest scale division

of the pressure measuring instrument exceed two times the specified accuracy.

6.3.2 PRESSURE MEASUREMENT LOCATIONS

Static pressure shall be determined on each side of the ATAHE both in the supply and exhaust air stream.

Pressure in the psychrometric chambers shall be measured for, and used in, humidity calculations.

Pressure measured at flow rate stations will be dependant on the type of flow measuring instruments used, (Section 6.4.1) but shall conform to accuracy limits stated in Section 6.3.1.

It is recommended that the pressure taps consist of 6.25mm (0.250 in) diameter nipples centred over 0.1mm (.004 in) diameter holes. The edges of these holes shall be free of burrs and other surface irregularities.

Static pressure locations shall consist of four pressure taps manifolded together through a piezometer ring with a common take-off to the pressure sensing instrument. For rectangular ducts these taps shall be centred vertically on the side panels and horizontally on the top and bottom panels. For round ducts the taps shall be equally spaced around the perimeter.

6.4 AIR FLOW MEASUREMENTS

Air flow rates at each of the measurement stations may be obtained by one of the following methods[49] provided the cumulative accuracy of the instrumentation will permit measurements within $\pm$ 2% of the actual flow rate.

(a) Venturi Tube

(b) Flow Nozzle

(c) Averaging Pitot Tubes

(d) Laminar Flow Element

Accuracy of the instrumentation shall be ascertained by pitot tube traverse measurements in accordance with AMCA Standard 210-67, Test Code for Air Moving Devices.

Air diffusers shall be placed before and after each air flow measurement device also in accordance with AMCA Standard 210-67. Diffusers shall be installed at least 5 duct diameters upstream of the measurement device and 2 duct diameters downstream of the measurement device.

The diameter for the duct section preceding and following the air flow measurement device shall be within $\pm \frac{1}{4}$ Ø of the specified duct diameter for a minimum of 8 times the nominal diameter preceding the device and 2 times the nominal diameter following the device.

6.5 PSYCHROMETRIC SAMPLING

Psychrometric sampling shall be measured before and after the heat exchanger, both in the supply and exhaust air streams (Figure 12).

To measure as accurate an average air condition as possible, a sampling grid shall be placed downstream of each air flow measuring station following the diffuser baffle. The sampling grid, depicted in Figure 15 provides for a minimum of four air samples from an equal matrix across the area of the air flow station. The grid shall be designed so that from any one sampling point the distance travelled to the outlet of the grid would be identical.

Wet bulb temperatures shall be read only under conditions which assure an air velocity of 3.5 to 10.0m/s over wet bulb and preferably near 5.0m/s, and only after sufficient time has been allowed for evaporative equilibrium to be

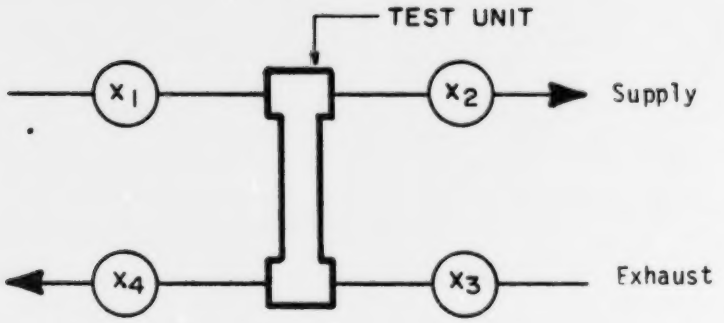


FIGURE 13 AIR FLOW THROUGH ATAHE FOR EFFECTIVENESS CALCULATIONS.

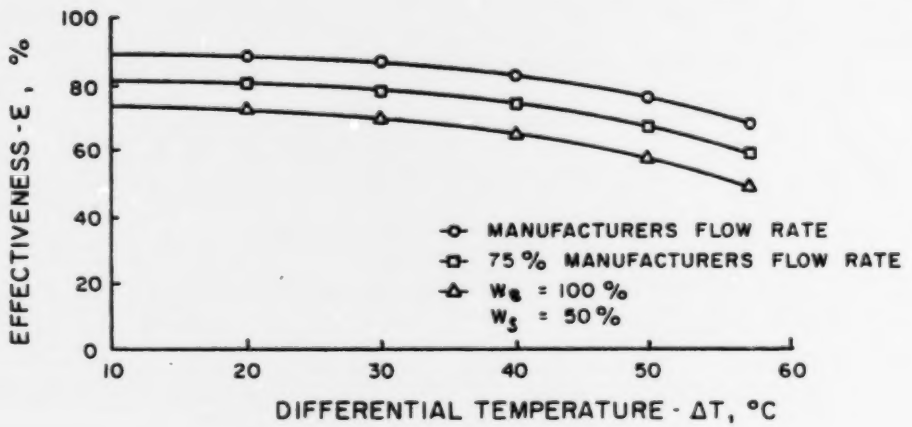


FIGURE 14 SAMPLE GRAPH - EFFECTIVENESS vs TEMPERATURE DIFFERENCE AND MASS FLOW RATE

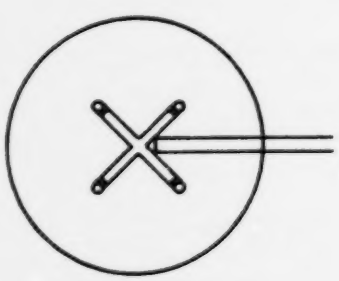


FIGURE 15 PYSCHROMETRIC SAMPLING GRID

attained. For wet bulb determinations at temperatures below freezing, a recommended method is to freeze a coating of ice approximately 0.05 centimeter thick directly on the bulb of the thermometer or other temperature sensing element. Another method is to preheat an air sample prior to temperature measurement, to bring the wet bulb temperature above 0°C. The preheat temperature should be as low as possible in order that the wet bulb depression be minimized. When an air sample is preheated a dry bulb measurement shall precede the heater and the dry and wet bulb temperatures read at the same station following the preheating. Humidity may also be determined through the direct measurement of dew point using basic apparatus providing an accuracy of $\pm$ 0.1°C.

6.6 TRACER GAS SAMPLING

A tracer gas injected into the air stream in conjunction with a gas chromatograph analyser or equivalent shall be used to determine cross flow contamination between the exhaust and the air intake streams.

The sampling grids used for psychrometric sampling shall be used for tracer gas sampling.

The gas chromatograph shall be capable of measuring the tracer gas to a minimum of 0.1% of the initial concentration.

6.7 ACOUSTIC MEASUREMENTS

Sound level measurement equipment should conform to ANSI Standard S1.4-1971, Specifications for Sound Level Meters, Type 2 General Purpose.

6.8 ELECTRICAL POWER MEASUREMENT

Electrical equipment associated with the normal operation of the ATAHE (i.e. fans, defrosting devices, etc.) shall be measured for electrical power consumption.

Instrumentation used for electrical power consumption measurements shall meet ANSI Standard C37.90a - 1974.

7.0 ATAHE TEST GUIDELINES

7.1 Pre Test Conditions

7.1.1 GENERAL

Prior to installation of the equipment to be tested, the ducts across the test section shall be connected and operated under the maximum negative pressure and flow rate that will be encountered under test conditions. Flow rates shall be determined at the four measurement stations. Any variation in flow rates greater than 3% shall be considered unacceptable.

The equipment to be tested shall be installed in the test section in accordance with the manufacturer's standard installation instructions.

The test duct, measuring equipment and the test unit shall be operated until equilibrium conditions are attained.

Air leakage into the ATAHE through the casing will improve the measured effectiveness of the unit, inverse to actual effectiveness in practical application. Casing leakage therefore shall be determined prior to effectiveness testing.

7.1.2 ATAHE Casing Leakage

The ATAHE test unit will be operated at 100% of the manufacturers specified load rating. Conditioning of air streams is not required during these tests.

With the ATAHE installed as received from the manufacturer, determine the mass flow rates at each of the measurement stations as indicated in Figure 12.

The casing shall then be sealed and the same test conditions repeated with mass flow rates at stations 1 and 3 matching exactly those ascertained previously.

The difference in mass flow rates at stations 2 and 4 from the rates measured in the previous test indicate casing leakage into the intake and exhaust air streams respectively.

Results shall be presented as a percentage of the leakage mass flow (ΔW_2 and ΔW_4) to that of the total mass flow (W_2 max and W_4 max respectively). Two percentages will be given, that of extraneous air entering the supply and extraneous air entering the exhaust.

If leakage is indicated, the ATAHE casing should remain sealed for the duration of the test program.

Casing leakage should not exceed 1% at the manufacturers rated load.

7.2 THERMAL EFFECTIVENESS

7.2.1 TEST SCHEDULE

The test schedule as shown in Table 11 shall be used for each of the following test conditions:

a) The manufacturers rated air flow through both the exhaust and supply sections of the ATAHE.

b) 75% of the rated flow through both sections of the ATAHE.

c) 100% rated flow through the exhaust section and 50% of the rated flow through the supply section.

Tests shall be performed with an exhaust-supply minimum humidity ratio difference of 0.003 kg/kg dry air. For the optional cooling mode tests, the minimum humidity ratio difference shall be 0.007 kg/kg dry air. Selected conditions shall be such that condensation will not occur in the exhaust air stream (intake air stream for optional cooling mode). It should be noted that this does not purport to represent real operating conditions.

The supply and exhaust air temperatures are to be held within - 0.5^{o}C and the humidity ratio within - 0.001 kg/kg dry air during a selected test phase.

TABLE 11. SAMPLE TEST SCHEDULE

Test No.	% Manufacturers' Flow Rate	Ambient Temperature °C (°F)	Supply Temperature °C (°F)	Exhaust Temperature °C (°F)
1	100	21 (69.8)	15 (59)	21 (69.8)
2	"	"	10 (50)	"
3	"	"	0 (32)	"
4 (i)	"	"	0 (32)	" @ RH $\geq$ 30%
5	"	"	-20 (- 4)	"
6 (i)	"	"	-20 (- 4)	" @ RH $\geq$ 30%
7	"	"	-40 (-40)	"
8	"	"	-40 (-40)	" @ RH $\geq$ 30%
9	75	"	-40 (-40)	"
10	"	"	-20 (- 4)	"
11	"	"	0 (32)	"
12	"	"	10 (50)	"
13	"	"	15 (59)	"
14	W_e = 100, W_s = 50	"	15 (59)	"
15	"	"	10 (50)	"
16	"	"	0 (32)	"
17	"	"	-20 (- 4)	"
18	"	"	-40 (-40)	"
19	100	10 (50)	-40 (-40)	"
20	"	"	-20 (- 4)	"
21	"	"	0 (32)	"
22	"	"	10 (50)	"
23	"	"	15 (59)	"
24 (ii)	100, P = 0	ambient	ambient	ambient
25 (ii)	100, P = +2.5Pa	"	"	"
26 (ii)	100, P = -2.5Pa	"	"	"
27 (iii)	100	24 (75.2)	30 (86)	24 (75.2)
28 (iii)	"	"	33 (91.4)	"
29 (iii)	"	"	36 (96.8)	"
30 (iii)	"	36 (96.8)	36 (96.8)	"
31 (iii)	"	33 (91.4)	33 (91.4)	"
32 (iii)	"	30 (86)	30 (86)	"

(i) Frosting and Icing Tests (ii) Supply Air Contamination Tests
(iii) Optional Summer Test Schedule.

Thermal effectiveness test shall be conducted with zero static pressure differential between measurement stations 2 and 3.

7.2.2 CALCULATION FOR THERMAL EFFECTIVENESS

The effectiveness of the heat exchanger is calculated as follows:

$$\varepsilon = \frac{W_s}{W_{min}} \quad \frac{X_2 - X_1}{X_3 - X_1}$$

where

ε = sensible, latent or total heat effectiveness

X = dry bulb temperature, humidity ratio or total enthalpy, respectively, at the locations indicated in Figure 12

W_s = mass flow rate of the supply, kilograms of dry air per unit of time (as measured at X_2)

W_e = mass flow rate of the exhaust, kilograms of dry air per unit of time (as measured at X_3)

W_{min} = minimum value of W_s or W_e

Total enthalpy is calculated as follows:

$$h = h_a + C_p X t + W (hg_o + C_{po} X t)$$

where

h = specific enthalpy in kJ per kg of dry air

h_a = specific enthalpy of dry air @ 0^oC = 17.88 $\frac{kJ}{kg}$

C_p = specific heat of dry air @ 0^oC = 1.006 $\frac{kJ}{kg\ ^oK}$

W = humidity ratio in $\frac{kg\ H_2O}{kg\ dry\ air}$

hg_o = specific enthalpy of saturated water vapour at 0^oC = 2501.3 $\frac{kJ}{kg}$

C_{po} = specific heat of saturated vapour @ 0^oC = 1.86 $\frac{kJ}{kg\ ^oK}$

t = temperature of air stream in ^oC

7.2.3 PRESENTATION OF RESULTS

The resultant effectiveness ε shall be shown in a family of curves, as shown in Figure 14, with effectiveness vs ΔT at the test conditions prescribed in Section 7.2.1.

7.3 ATAHE PRESSURE DROP AND DEFROSTING/DE-ICING EFFECTIVENESS

Air pressure drop across the ATAHE shall be measured at manufacturers' flow rates on both the exhaust and supply air streams.

Data shall be taken at each test point prescribed in

Section 7.2.1 under the above flow conditions.

The pressure drop through each section will be calculated from the following equations:

$$\Delta P_s = P_5 - P_6$$

$$\Delta P_e = P_7 - P_8$$

where

ΔP_s = pressure drop through supply section of ATAHE

ΔP_e = pressure drop through exhaust section of ATAHE

P_5 = static pressure at supply inlet of ATAHE

P_6 = static pressure at supply outlet of ATAHE

P_7 = static pressure at exhaust inlet of ATAHE

P_8 = static pressure at exhuast outlet of ATAHE.

Results shall be shown in graphical form with ΔP versus flowrate.

Frosting will be indicated by an increase in pressure drop across the ATAHE during those tests (Section 7.2.1) which would promote such conditions.[48] Defrosting effectiveness shall be determined by the decrease in ΔP and the time required for the ΔP to return to normal.

A definition will be required of the maximum acceptance value of AP_e, or percentage drop in air flow before defrosting takes place.

Icing and water drainage capability will be determined by a visual inspection of the ATAHE equipment after the frosting tests.

7.4 SUPPLY AIR CONTAMINATION

7.4.1 PROCEDURE

An inert tracer gas is injected into a turbulent region between stations 2 and 3 (before fan #1). Air samples are then drawn from stations 2 and 3 to determine the cross contamination. A sample will also be drawn from station 1 to confirm the purity of fresh air.

The injection rate of inert gas shall be sufficient that a carryover of 0.1% will be within the measuring capability of the gas chromatograph being used.

A continuous air stream shall be drawn from each sampling grid identified above. Samples shall be drawn from these streams by laboratory approved sampling procedures.

No heat or moisture shall be added to the system between stations 2 and 3 during tracer gas testing. Manufacturers' air flow rates shall be used with pressure differences between stations 2 and 3 of -2.5, 0.0 and 2.5 pascal.

7.4.2 CALCULATION AND PRESENTATION OF TEST RESULTS

The cross contamination is equal to the tracer gas concentration in the supply air at station 2 divided by the tracer gas concentration in the exhaust air at station 3.

The results of the tracer gas tests shall be presented as a percentage of the concentration existing at station 2 to that measured at station 3.

7.5 ACOUSTIC MEASUREMENTS

Acoustical measurements shall be made with the ATAHE and associated fans and other equipment operating at 100% of design capacity.

The sound pressure levels will be measured in dBA re 0.0002 microbar and in octave bands of 31, 63, 125, 250, 500, 1000, 2000, 4000 and 8000 Hz centre frequencies.

At least 4 sets of noise level readings shall be recorded, where each set is taken at a 3 foot distance from each of the four principal orthogonal surfaces of the equipment.

For all noise tests, the ambient noise level of the test

area should be at least 10dB below the equipment noise levels.

7.6 ELECTRICAL POWER CONSUMPTION

Electrical consumption measurements shall be taken with all equipment under design load conditions.

Fans shall be operated at rated flow rates with normal air pressure drop either as stated by the manufacturer or ASHRAE Handbook of Fundamentals, Chapter on Air Duct Design, 1977.

Defrosting and deicing mechanisms, if electrical, shall be activated after air moving equipment consumptions are ascertained.

Electrical power consumption results shall be presented for air moving equipment and also for accessory equipment in Kw.

Dependant on ATAHE design the defrosting/deicing phase

may occur with reduced air circulation rates in which case the results will be given for 100% rated flow and for subsequent reduced flow rates.

7.7 MAINTENANCE REQUIREMENTS

The manufacturer's maintenance program should be reviewed to ensure that any maintenance procedures are appropriate for the intended installations.

8.0 ORGANIZATIONS CAPABLE OF UNDERTAKING PERFORMANCE TESTING

A questionnaire survey was conducted to determine appropriate organizations which have the expertise, resources and facilities to evaluate the performance of ATAHE's in the residential capacity range.

The intermediate objectives of this survey were identified as the following:

- determine organizations interested in the performance testing of residential ATAHE's
- identify organizations presently testing thermal energy recovery equipment
- identify organizations presently testing ATAHE's and the types of flow capacities being tested
- determine extent of tests conducted on ATAHE's
- identify lower temperature limit of test facility
- determine organization's willingness to expand facilities to include R & D activities and testing of ATAHE's in the residential capacity range
- identify costs for the performance testing of a unit residential ATAHE.

Appendix III includes a sample copy of the questionnaire sent to various organizations within the following sectors:

- non profit Research Institutions
- Universities
- Commercial Testing Laboratories
- Industrial Laboratories

A mailing list of these organizations is also included in Table 13.

Costs on the performance testing of residential ATAHE's were submitted directly from the organization to the Client, Canada Mortgage and Housing Corporation, because of the confidentiality of the information.

A summary of the responses to the questionnaire is shown in Table 12.

Information gathered from this survey will be used by other authorities, who will determine the viability and nature of a performance testing programme.

TABLE 12

Summary of Responses to
Questionnaire on Testing Facilities

Number of Enquiries		37		
Responses to General Enquiry			Cost Quoted	Cost Not Quoted
Canadian	Interested	13	6	7
	Not Interested	10	-	-
U.S.	Interested	3	1	2
	Not Interested	1	-	-
Total Responses		27		

TABLE 13 MAILING LIST FOR QUESTIONNAIRE

1. Already testing ATAHE Y-Yes; N-No; SE-Some Equipment; - indicates no respons
2. Intested in carrying out tests Y-Yes; N-No; MB-maybe
3. Approximate cost of testing given Y-Yes; N-No

		1	2	3
1.	National Research Council Division of Building Research Montreal Road OTTAWA, Ontario K1A OR6 Attn: Energy & Services (613) 993-1421	N	N	-
2.	Concordia University Sir George Williams Campus 1455 de Maissonneuve Blvd. W. MONTREAL, Quebec H3G 1M8 Attn: Dr. M.N.S. Swamy Dean of Engineering Room H907-7 (514) 879-5995	Y	Y	Y
3.	Canadian Gas Association 55 Scarsdale Rd. DON MILLS, Ontario. M3B 2R3 Attn: John Overall. (416) 447-6465	N	N	-
4.	Kansas State University Engineering Experiment Station Seaton Hall MANHATTAN, Kansas, 66506	-	-	-
5.	University of Missouri - Rolla ROLLA, Missouri U.S.A. 65401 Attn: Dr. Ronald H. Howell Mechanical Engineering. (314) 341-4161	Y	Y	Y
6.	University of Manitoba WINNIPEG, Manitoba R3T 2N2 Attn: Professor Chant Department of Mechanical Engineering (204) 474-9463 474-8880	Y	Y	Y
7.	University of Saskatchewan SASKATOON, Saskatchewan S7N OWO Attn: Dr. R. Besant (306) 343-3101	Y	N	-

		1	2	3
8.	University of Toronto Faculty Office Applied Science and Engineering Galbraith Bldg. 35 St. George Street TORONTO, Ontario. M5S 1A4 Attn: Dean Slemon (416) 978-3134	N	Y	-
9.	University of Alberta EDMONTON, Alberta T6G 2E5 Attn: Dr. Bellows Head of Mechanical Engineering (403) 432-3111 Prof. Sadler (403) 432-3450	N	N	-
10.	Ohio State University Building Research Laboratory 2070 Neil Ave. COLUMBUS, Ohio 43210 (614) 422-4931	-	-	-
11.	Texas A & M University Engineering Test Laboratory College Station, TEXAS 77843 Attn: Charles E. Neeley (713) 779-3880 Ext. 217	N	Y	N
12.	Lawrence Berkely Laboratory 1 Cyclotron Road BERKELEY, Calif. 94720 Attn: W. Fisk Energy & Environment Department (415) 486-5910	Y	Y	N
13.	Ryerson Polytechnical Institute 50 Gould St. TORONTO, Ontario. M5B 1E8 Attn: Mr. T.E. Wisz Dean of Technology (416) 595-5000	-	-	-
14.	Underwriters Laboratories Inc. 333 Pfingsten Rd. NORTHBROOK, Ill. 60062 (312) 272-8800	N	N	-
15.	Canadian Standards Assoc. 178 RExdale Blvd. REXDALE, Ontario. M9W 1R3 (416) 744-4000	-	-	-

	1	2
16. Warnoch Hersey Professional Services Ltd. 250 Madison Street TORONTO, Ontario. (416) 924-9691	N	Y
17. St. Clair College of Applied Arts & Technology 2000 Talbot Road W. WINDSOR, Ontario. N9A 6S4 (519) 966-1656 Attn: Roger Baxter Technical Arts & Trades	N	N
18. Ontario Hydro Research Laboratory 800 Kipling Avenue TORONTO, Ontario. Attn. John Jackson (416) 231-4111	Y	Y
19. University of Windsor WINDSOR, Ontario. N9B 3P4 Attn: Cameron MacInnes Dean, Faculty of Engineering (519) 253-4232	N	N
20. University of Waterloo WATERLOO, Ontario. N2L 3G1 Attn: Dr. K.G.T. Hollands Faculty of Engineering (519) 885-1211	Y	N
21. McGill University 845 Sherbrooke Street West MONTREAL, Quebec H3A 2T5 Attn: Dean, Faculty of Engineering (514) 392-4311	N	Y
22. Queens University at Kingston KINGSTON, Ontario. K7L 3N6 Attn: Dean, Faculty of Engineering (613) 547-5511	N	N

	1	2	3
23. Technical University of Nova Scotia P.O. Box 1000 HALIFAX, Nova Scotia B3J 2X4 Attn: Dr. Warner, Head, Mechanical Engineering Department (902) 429-8300	-	-	-
24. Memorial University of Newfoundland Elizabeth Avenue ST. JOHN'S, Newfoundland A1C 5S7 Attn: Head, Mechanical Engineering Department (709) 737-8000	-	-	-
25. University of British Columbia 2075 Weslrook Mall VANCOUVER, B.C. V6T 1W5 Attn: Dean, Faculty of Engineering. (604) 228-2211	-	-	-
26. University of New Brunswick Fredericton Campus P.O. Box 4400 FREDERICTON, N.B. E3E 5A3 Attn: Dr. D. Bonham Head, Mechanical Engineering Department. (506) 453-4995	N	N	-
27. University of Prince Edward Island CHARLOTTETOWN, P.E.I. C1A 4P3 Attn: Dean, Faculty of Engineering. (902) 892-4121	-	-	-
28. Battelle Memorial Institute 505 King Ave. COLUMBUS, Ohio 43201 (614) 424-6424	-	-	-
29. Fanshawe College of Applied Arts and Technology P.O. Box 4005 LONDON, Ontario. N5W 5H1 Attn: Mr. P.N. Brown Dean of Technology (519) 452-4405	N	Y	-

		1	2	3
30.	Nova Scotia Research Foundation 100 Fenwick St., Box 790 DARTMOUTH, Nova Scotia B2Y 3Z7 Attn: Dr. J.E. Blanchard	N	Y	-
31.	Research Council of Alberta Industrial and Engineering Services Dept. 11315 - 87th Avenue EDMONTON, Alberta T6G 2C2 Attn: J. Holub	N	N	-
32.	Saskatchewan Research Council University of Saskatchewan 30 Campus Drive SASKATOON, Saskatchewan S7N 0X1 Attn: W.H.W. Husband	Y	Y	Y
33.	British Columbia Research Council Division of Engineering 3650 Westbrook Mall VANCOUVER, B.C. V6S 2L2 Attn: F.S. Fraser	N	Y	-
34.	Hydro-Quebec Institut de Recherches de l'Hydro Quebec (IREQ) 1800 Mtee Ste Julie Vrns VARENNES, Quebec JOL 2PO	-	-	-
35.	Research and Productivity Council College Hill Road FREDERICTON, New Brunswick E3B 4J7	N	Y	Y
36.	University of Western Ontario Faculty of Engineering LONDON, Ontario. Attn: John Franasuk.	Y	Y	N
37.	Ontario Research Foundation Sheridan Park Research Community Mississauga, Ontario L5K 1B3	Y	Y	Y

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APPENDIX I

Air-to-Air Heat Exchanger Availability Study

Questionnaire and List of Companies

THE ECE GROUP Consulting Engineers 205 Lesmill Road Don Mills Ontario M3B 2V1 (416) 449 1030
Corporate Partners: Jack Chisvin & Associates Limited, G. Granek Limited, CME Engineers Limited

February 13, 1981

Dear Sir:

The ECE Group, in co-operation with the Ontario Research Foundation, has been commissioned to undertake a study on Air-to-Air Heat Exchangers for residential use in Canada, on behalf of the Canada Mortgage and Housing Corporation, and the Ministry of Energy Mines and Resources. The study will involve two phases. The first will be investigatory. The second will involve the physical testing of air-to-air heat exchangers (ATAHE).

It is in reference to the first phase that I am presently writing to you. I am attempting to provide a review of ATAHEs which are manufactured in Canada, or available for importation.

The heat exchangers we are primarily interested in are heat pipes, plate heat exchangers and heat wheels.

I would appreciate receiving any information you may have relating to ATAHEs in the above categories.

Naturally, since we are concerned with heat exchangers for residential dwellings, we can expect ventilation rates as low as 100 cfm. Since most existing ATAHEs are built with larger flow rates in mind, any information regarding ATAHEs presently being developed for this lower flow rate would be of greatest interest.

I would also appreciate if you could give me an estimate of the installed cost of such systems in Canada.

...2

If it is not too much trouble, please fill in the enclosed questionnaire, and return it with any additional information or related material you may have.

I look forward to hearing from you.

Yours very truly,

THE ECE GROUP

Michael Martin,
Junior Engineer

MM/dhp
Encl.

AIR-TO-AIR HEAT EXCHANGER SURVEY

Send to: Michael Martin
The ECE Group
205 Lesmill Road
Don Mills, Ontario
CANADA
M3B 2V1

Type of ATAHEs available:

a) Heat Pipe ☐
b) Plate Exchanger ☐
c) Heat Wheel ☐
d) Coil Loop ☐
e) Thermosiphon ☐
f) Other _____

i) Manufactured in Canada ☐
ii) Available for importation ☐
iii) Presently in an advanced stage of development and which could be made available in Canada. ☐

Characteristics of ATAHEs

a) Approx. cfm range _____
b) Approx. cost (installed) $_____
c) Maintenance required _____
d) Adaptable to existing ventilation systems Yes ☐ No ☐
e) Approx. dimensions _____

MM/dhp
13 Feb 81
S-3608

A. The Air Changer Co. Ltd.
334 King Street East
Studio 505
Toronto, Ontario
M5A 1K8

B. Henlex Inc.
1300 Boul. Jules-Poitras
Montreal, Quebec
H4N 1X8

C. Z-Air Fabrication Inc.
690 Place Trans Canada
Longueul, Quebec
J4G 1P2

D. D.C. Heat Exchangers
R. R. #3
Saskatoon, Saskatchewan

E. Conservation Energy Systems
(VanEE Air Exchangers)
Box 8280
Saskatoon, Saskatchewan
S7K 6C6

1. Alpha-Laval
113 Park Street South
Peterborough, Ontario
K9J 3R8

2. Indusco Sales
48 Chauncey Avenue
Toronto, Ontario
M8Z 2Z4

3. Heat Reclaim Systems
2000 Sheppard Avenue W.
Suite 204
Downsview, Ontario
M3N 1Z2

4. APV-Crepaco
1250 Ormont Drive
Weston, Ontario
M9L 2V4

5. Temprite Industries Ltd.
6470 Viscount Road
Mississauga, Ontario
L4V 1H3

6. Tranter Canada Ltd.
6700 Finch Avenue West
Rexdale, Ontario
M9W 5P5

7. Berner Internaional Corp.
P. O. Box 5205
New Castle, PA.
16105, U.S.A.

8. Flakt Canada Limited
55 University Avenue
Suite 600
Toronto, Ontario
M5J 2H7

9. McCarthy & Robinson
(1966) Limited
321 Progress Avenue
Scarborough, Ontario
M1P 2Z7

10. The Wing Co.
Division of Wing
Industries Inc.
125 Moen Avenue
Cranford, N.J.
07016, U.S.A.

11. Bry-Therm Division
Bry-Air, Inc.
P. O. Box 795
Sunbury, Ohio
43074, U. S. A.

12. P. M. Wright Ltd.
1111 Finch Avenue W.
Downsview, Ontario
M3J 2E5

13. Isothermics Inc.
P. O. Box 86
Augusta, N. J.
07822, U.S.A.

14. Keep-Rite Products Ltd.
44 Elgin Street
Brantford, Ontario
N3T 5P4

15. Longhill Energy Products Ltd.
293 Markham Road
Unit 6
Richmond Hill, Ontario
L4C 1J2

16. Chromalox Canada Inc.
1200 Aerowood Drive
Unit 42
Mississaua, Ontario
L4W 2S7

17. Blanchard - Bohn Inc.
1415 Lawrence Ave. W.
Suite 3
Toronto, Ontario
M6L 1A9

18. Carrier Ontario Distributors
9 Cityview Drive
Rexdale, Ontario
M9W 5A5

19. Canadian Command-Aire Ltd.
250 Consumers Road
Willowdale, Ontario
M2J 4V6

20. Climate Master
8052 Torbram Road
Bramalea, Ontario
L6T 3T2

21. Eng. A. Engineered Air
282 Belfield Road
Rexdale, Ontario
M9W 1H5

22. En-Mar Systems Ltd.
1297 Industrial Road
Cambridge, Ontario
M3H 4W3

23. Airquin Systems Inc.
25 Kelfield Street
Rexdale, Ontario
M9W 5A1

24. Ross & McNeilage Ltd.
Manufacturers Reps.
85 Ellesmere Road
Suite 232
Scarborough, Ontario
M1R 4B9

25. Honeywell Limited
300 Yorkland Blvd.
Willowdale, Ontario
M2J 1S1

26. Keep-Rite Products Ltd.
150 Consumers Road
Suite 302
Willowdale, Ontario
M2J 1P9

27. Lennox Industries (Canada) Ltd.
400 Norris Glen Road
Etobicoke, Ontario
M9C 1H5

28. J. W. Swanson Ltd.
3171 Lenworth Drive
Mississauga, Ontario
L4X 2G6

29. Michael Stuart Co. Ltd.
Engineering Specialties
300 Steelcase Road W.
Unit 7
Markham, Ontario
L3R 1B3

30. Preston, Phipps Equipment
Limited
48 Shaft Road
Rexdale, Ontario
M9W 4M2

31. K. D. Huhn Ltd.
168A Oakville Road
Downsview, Ontario
M3N 1W1

32. Remington Air
Conditioning Ltd.
48 Chauncey Avenue
Toronto, Ontario
M8Z 2Z4

33. Trane Company of
Canada, Ltd.
4051 Gordon Baker Rd.
Scarborough, Ontario
M1W 2P3

34. United Air Specialists
(Canada) Ltd.
115 Bowes Road
Thornhill, Ontario

35. Westinghouse Canada Ltd.
Air Conditioning Dept.
492 Speers Road
Oakville, ontario
L6K 2G3

36. York Air Conditioning
and Refrigeration
Div. of Borg-Warner
(Canada) Ltd.
1188 Martingrove Road
Rexdale, Ontario
M9W 5M9

37. B. D. Wait Co. Ltd.
430 Wyecroft Road
Oakville, Ontario
L6K 2G9

38. Air Dynamics Inc.
1918 Potrero Avenue
P. O. Box 3666
South El Monte, CA.
91733, U. S. A.

39. Air Energy Recovery
Co., Inc.
5952 Royal Lane
Suite 254
Dallas, Texas
75230, U. S. A.

40. Air Enterprises Inc.
735 Seiberling Street
Akron, Ohio
44306, U. S. A.

41. Airtherm Mfg. Co.
9339 Dielman Indl. Ct.
P. O. Box 7039
St. Louis, MO.
63132, U. S. A.

42. American Standard Heat
Transfer Division
P. O. Box 1102
Buffalo, N.Y.
14240, U. S. A.

43. Ammerman Company
a Sub. of General
Resource Corp.
201 S. 3rd Street
Hopkins, MN
55343, U. S. A.

44. Applied Air Systems Inc.
2475 Doswell Avenue
St. Paul, MN.
55108, U. S. A.

45. Armstrong Machine Works
816 Maple Street
Three Rivers, MI.
49093, U. S. A.

46. Bahnson, Industrial Air
Quality Div.
Envirotech Corp.
1001 S. Marshall St.
Winston-Salem, NC.
27108, U. S. A.

47. Bartell Wagner Div.
National Standard Co.
601 N. 8th Street
Miles, MI.
49120, U. S. A.

48. Berdell Industries, Inc.
8-16 43rd Avenue
Long Island City, N.Y.
11101, U. S. A.

49. Blazer Industries Inc.
700 21st Street
Paterson, N.J.
07513, U. S. A.

50. Buffalo Forge Co.
490 Broadway
Buffalo, N.Y.
14240, U. S. A.

51. E.K. Campbell Co.
1890 Manchester Trfkwy.
Kansas City, MO.
64126, U. S. A.

52. Conseco, Inc.
611 North Road
Medford, WI
54451, U. S. A.

53. Corning Glass Works
Advanced Engine Components
Erwin Plant
Corning, New York
14830, U. S. A.

54. Cylcotherm Division
Oswego Package Boiler
Co., Inc.
157 E. First Street
Oswego, N. Y.
13126, U. S. A.

55. Des Champs Laboratories Inc.
Box 348
Dorine Industrial Park
East Hanover, NJ.
07936, U. S. A.

56. Dravo/Hastings
One Oliver Plaza
Pittsburgh, PA.
15222, U. S. A.

57. Dunham-Bush, Inc.
175 South Street
West Hartford, CT.
06110, U. S. A.

58. Eclipse Inc.
1665 Elmwood Road
Rockford, IL.
61103, U. S. A.

59. Eclipse Lookout Co.
P. O. Box 4756
Chattanooa, TN.
37405, U. S. A.

60. Econo-Therm Energy Sys.
11535 K-Tel Drive
Minnetonka, MN.
55343, U. S. A.

61. Energy Recovery Co.
N112 W18700 Nequon Rd.
P. O. Box 158
Germantown, WI.
53022, U. S. A.

62. Energy Recovery Systems Inc.
P. O. Box 233
Lincroft, NJ.
07738, U. S. A.

63. Energy Systems Inc.
Five Lawrene Street
Bloomfield, NJ.
07003, U. S. A.

64. Exothermics Corp.
828 Prouty Avenue
Toledo, Ohio
43609, U. S. A.

65. General Resource Corp.
201 S. 3rd Street
Hopkins, MN.
55343, U. S. A.

66. Governair Corp.
4840 N. Sewell
Oklahoma City, OK.
73118, U. S. A.

67. Halco Products Co.
Div. of Hale Engineering Co. Inc.
100 N. Gordon Street
Elk Grove Village, IL.
60007, U. S. A.

68. Hirt Combustion Engineers
931 S. Maple Avenue
Montebello, CA.
90640, U. S. A.

69. Jackson & Church Div,
York-Shipley, Inc.
391 Greendale Road
York, PA
17405, U. S. A.

70. The Kanthal Corp.
Wooster Street
Bethel, CT.
06801, U. S. A.

71. Miller-Picking Corp.
P. O. Box 130
Johnstown, PA.
15907, U. S. A.

72. Perfectair Equipment Co.
52 First Street
Cambridge, MA.
02141, U. S. A.

73. Perfex Group
McQuay-Perfex, Inc.
500 W. Oklahoma Ave.
Milwaukee, WI.
53207, U. S. A.

74. Rotamist Division
Lake Erie Technology Inc.
210 French Road
Buffalo, N. Y.
14227, U. S. A.

75. Rupp Industries Inc.
11550 Rupp Drive
Burnsville, MN.
55337, U. S. A.

76. Thermal Devices Inc.
P. O. Box 151
Slag Road
Mayville, WI.
53050, U. S. A.

77. Thermal Energy Storage Inc.
10635A Roselle Street
Suite A
San Diego, CA
92121, U. S. A.

78. Thermo-Kinetics Industries
716 E. Fairfield Road
Greenville, SC
29605, U. S. A.

79. Tjernlund Customair Inc.
2140 Kasota Avenue
St. Paul, MN.
55108, U. S. A.

80. United Air Specialists Inc.
4440 Creek Road
Cinncinnati, Ohio
45242, U. S. A.

81. United Aircraft Products Inc.
Dayton Inernational Airport
P. O. Box 1335
Dayton, Ohio
45401, U. S. A.

82. Webco Engineering Co.
4535 W. Chestnut Expressway
Springfield, MO.
65802, U. S. A.

83. Melco Sales, Inc.
3030 East Victoria St.
Compton, California
90221, U. S. A.

84. Advanced Idea Mechanics Limited
26 Ski Valley Cres.
London, Ontario
N6K 3H3

85. R.L. Fulton Equipment Limited
P. O. Box 1402 N
Halifax, Nova Scotia
B3K 5H7

86. Genvex Energiteknik A/S
Blegdamsvej 104
DK-2100
KOBENHAVN O
Denmark

87. Norlett
N-1801
Askim
Norway

88. Acoustics & Envirometrics Limited
Ruxley Towers
Claygate
Surrey KT10 OUF
England

89. Enercon Industries Ltd.
2073 Cornwall Street
Regina, Saskatchewan
S4P 2K6

90. Advanced Idea Mechanics
P. O. Box 48
Hyde Park, Ontario
NOM 1ZO

91. Passive Solar Products Limited
2234 Hanselamn Ave.
Saskatoon, Saskatchewan
S7L 6A4

92. Atlantic Air Seal (Fredericton)
880 Hanwell Place,
Hanwell Road
Fredericton, N. B.
E3B 2R7

93. Q-Dot Corp.
151 Regal Row
Suite 120
Dallas, Texas
75247, U. S. A.

94. Regenerative Heat Corp.
P. O. Box 78
Middletown, Conn.
06457, U. S. A.

95. Howden Heat Pipe Products
75 Mill Street
Newton, N.J.
07860, U. S. A.

96. Flo-Con Corp.
520 North Avenue
Garwood, N.J.
07027, U. S. A.

97. Hunt & Moscrop (Canada) Inc.
P. O. Box 355
Granby, Quebec
J2G 8E5

98. McQuay Corp.
McQuay-Perfex Inc.
P. O. Box 1551
Minneapolis, Minn.
55440, U. S. A.

Additional Companies Manufacturing Residential Size Units but not Solicited for Detailed Information

BAHCO VENTILATION	Bahco Ventilation Ltd. Bahco House Beaumont Road Banbury OX167TB U.K.
FAN-X-CHANGER	P.M. Wright Ltd. 1300 Jules-Poitras, St. Laurent Montreal, P.Q. (514) 337-3331
CAPELATOR	Advance Energy Technologies Inc. P.O. Box 387, Clifton Park New York 12065 U.S.A. (518) 371-2140
ZERO ENERGY PRODUCTS LTD. (for Mitsubishi)	2218 Avenue C North Saskatoon, Saskatchewan S7L 6C3
THE STAR HEAT EXCHANGER CORPORATION (Development only as of Dec. 1981)	1041 Buoy Drive Coquitlam B.C. V3H 3M3 (604) 936-3210

APPENDIX II

Results of Testing

RESULTS OF TESTS ON PLATE TYPE AIR-TO-AIR HEAT EXCHANGERS

Heat Exchanger	Type	Wt. (kg.)	Cost ($)	Accessories	Effectiveness*	Airstream Static Pressure* (mm H_2O)	Fan Performance Test*	
							Pressure (mm H_2O)	Power (W)
VMC Genvex	crossflow (Al plates)	68	n.a.	2 fans 2 filters 220V 2 speed fan motor	0.57	n.a.	17.78 12.45	137 (high) 118 (low)
Flakt RDAA	crossflow (Al plates)	36	850	2 fans 2 Filters 220 V 2 speed fan motor preheating element	0.65	6.60	16.76	145 (high)
Plastic Sheet	counterflow	63	n.a.	self built	0.48	2.54	no fans	
Aldes VMPI	counter & crossflow	45	2500	2 fans ducting diffuser	0.68	3.05	n.a.	
Des Champs 74	counterflow	31	250	2 fans (external)	0.70	4.83	n.a.	

* All values for operation at 47 L/s (100 cfm).

Source: Performance af Residential Air-to-Air Heat Exchangers: Test Results and Methods (36).

APPENDIX III

Results of Enquiring for Carrying Out Testing

MAILING LIST FOR QUESTIONNAIRE

1. Already testing ATAHE Y-Yes; N-No; SE-Some Equipment; - indicates no response
2. Intested in carrying out tests Y-Yes; N-No; MB-maybe
3. Approximate cost of testing given Y-Yes; N-No

		1	2	3
1.	National Research Council Division of Building Research Montreal Road OTTAWA, Ontario K1A OR6 Attn: Energy & Services (613) 993-1421	N	N	-
2.	Concordia University Sir George Williams Campus 1455 de Maissonneuve Blvd. W. MONTREAL, Quebec H3G 1M8 Attn: Dr. M.N.S. Swamy Dean of Engineering Room H907-7 (514) 879-5995	Y	Y	Y
3.	Canadian Gas Association 55 Scarsdale Rd. DON MILLS, Ontario. M3B 2R3 Attn: John Overall. (416) 447-6465	N	N	-
4.	Kansas State University Engineering Experiment Station Seaton Hall MANHATTAN, Kansas, 66506	-	-	-
5.	University of Missouri - Rolla ROLLA, Missouri U.S.A. 65401 Attn: Dr. Ronald H. Howell Mechanical Engineering. (314) 341-4161	Y	Y	Y -
6.	University of Manitoba WINNIPEG, Manitoba R3T 2N2 Attn: Professor Chant Department of Mechanical Engineering (204) 474-9463 474-8880	Y	Y	Y
7.	University of Saskatchewan SASKATOON, Saskatchewan S7N OWO Attn: Dr. R. Besant (306) 343-3101	Y	N	-

		1	2	3
8.	University of Toronto Faculty Office Applied Science and Engineering Galbraith Bldg. 35 St. George Street TORONTO, Ontario. M5S 1A4 Attn: Dean Slemon (416) 978-3134	N	Y	-
9.	University of Alberta EDMONTON, Alberta T6G 2E5 Attn: Dr. Bellows Head of Mechanical Engineering (403) 432-3111 Prof. Sadler (403) 432-3450	N	N	-
10.	Ohio State University Building Research Laboratory 2070 Neil Ave. COLUMBUS, Ohio 43210 (614) 422-4931	-	-	-
11.	Texas A & M University Engineering Test Laboratory College Station, TEXAS 77843 Attn: Charles E. Neeley (713) 779-3880 Ext. 217	N	Y	N
12.	Lawrence Berkely Laboratory 1 Cyclotron Road BERKELEY, Calif. 94720 Attn: W. Fisk Energy & Environment Department (415) 486-5910	Y	Y	N
13.	Ryerson Polytechnical Institute 50 Gould St. TORONTO, Ontario. M5B 1E8 Attn: Mr. T.E. Wisz Dean of Technology (416) 595-5000	-	-	-
14.	Underwriters Laboratories Inc. 333 Pfingsten Rd. NORTHBROOK, Ill. 60062 (312) 272-8800	N	N	-
15.	Canadian Standards Assoc. 178 RExdale Blvd. REXDALE, Ontario. M9W 1R3 (416) 744-4000	-	-	-

	1	2	3
16. Warnoch Hersey Professional Services Ltd. 250 Madison Street TORONTO, Ontario. (416) 924-9691	N	Y	-
17. St. Clair College of Applied Arts & Technology 2000 Talbot Road W. WINDSOR, Ontario. N9A 6S4 (519) 966-1656 Attn: Roger Baxter Technical Arts & Trades	N	N	-
18. Ontario Hydro Research Laboratory 800 Kipling Avenue TORONTO, Ontario. Attn. John Jackson (416) 231-4111	Y	Y	N
19. University of Windsor WINDSOR, Ontario. N9B 3P4 Attn: Cameron MacInnes Dean, Faculty of Engineering (519) 253-4232	N	N	-
20. University of Waterloo WATERLOO, Ontario. N2L 3G1 Attn: Dr. K.G.T. Hollands Faculty of Engineering (519) 885-1211	Y	N	-
21. McGill University 845 Sherbrooke Street West MONTREAL, Quebec H3A 2T5 Attn: Dean, Faculty of Engineering (514) 392-4311	N	Y	Y -
22. Queens University at Kingston KINGSTON, Ontario. K7L 3N6 Attn: Dean, Faculty of Engineering (613) 547-5511	N	N	-

		1	2	3
23.	Technical University of Nova Scotia P.O. Box 1000 HALIFAX, Nova Scotia B3J 2X4 Attn: Dr. Warner, Head, Mechanical Engineering Department (902) 429-8300	-	-	-
24.	Memorial University of Newfoundland Elizabeth Avenue ST. JOHN'S, Newfoundland A1C 5S7 Attn: Head, Mechanical Engineering Department (709) 737-8000	-	-	-
25.	University of British Columbia 2075 Weslrook Mall VANCOUVER, B.C. V6T 1W5 Attn: Dean, Faculty of Engineering. (604) 228-2211	-	-	-
26.	University of New Brunswick Fredericton Campus P.O. Box 4400 FREDERICTON, N.B. E3E 5A3 Attn: Dr. D. Bonham Head, Mechanical Engineering Department. (506) 453-4995	N	N	-
27.	University of Prince Edward Island CHARLOTTETOWN, P.E.I. C1A 4P3 Attn: Dean, Faculty of Engineering. (902) 892-4121	-	-	-
28.	Battelle Memorial Institute 505 King Ave. COLUMBUS, Ohio 43201 (614) 424-6424	-	-	-
29.	Fanshawe College of Applied Arts and Technology P.O. Box 4005 LONDON, Ontario. N5W 5H1 Attn: Mr. P.N. Brown Dean of Technology (519) 452-4405	N	Y	-

		1	2	3
30.	Nova Scotia Research Foundation 100 Fenwick St., Box 790 DARTMOUTH, Nova Scotia B2Y 3Z7 Attn: Dr. J.E. Blanchard	N	Y	-
31.	Research Council of Alberta Industrial and Engineering Services Dept. 11315 - 87th Avenue EDMONTON, Alberta T6G 2C2 Attn: J. Holub	N	N	-
32.	Saskatchewan Research Council University of Saskatchewan 30 Campus Drive SASKATOON, Saskatchewan S7N 0X1 Attn: W.H.W. Husband	Y	Y	Y
33.	British Columbia Research Council Division of Engineering 3650 Westbrook Mall VANCOUVER, B.C. V6S 2L2 Attn: F.S. Fraser	N	Y	-
34.	Hydro-Quebec Institut de Recherches de l'Hydro Quebec (IREQ) 1800 Mtee Ste Julie Vrns VARENNES, Quebec JOL 2P0	-	-	-
35.	Research and Productivity Council College Hill Road FREDERICTON, New Brunswick E3B 4J7	N	Y	Y
36.	University of Western Ontario Faculty of Engineering LONDON, Ontario. Attn: John Franasuk.	Y	Y	N
37.	Ontario Research Foundation Sheridan Park Research Community Mississauga, Ontario L5K 1B3	Y	Y	Y